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▼

Destructive Complex

WHENEVER we hear that John J. Doe, product engineer, has been made vice-president in charge of engineering of his company, our casehardened old editorial heart skips a couple of beats. We know that at least one company has come, or has been forced, to the realization that the engineer must be recognized—and listened to—if industry is to endure and thrive. We know that one company is going to have engineering guidance in its executive functioning.

We used to feel, when an engineer had been thus recognized, that the least we could do was to make mention of the fact in the "Engineers and Their Activities" department. But, believe it or not, such is the "professional modesty" of the typical engineer that when we would write him for a photograph to go with an item, his almost invariable reply would be a polite "Thank you, but no. I do not want to appear to be seeking publicity. Our company is funny about such things and I am sure that you will understand. . . ." Hence the late, lamented and untimely demise of the department.

This same professional modesty crops up in another way. In the course of editorial visits and visitations we frequently come upon a fine new engineering design. Instinctively we feel that a thorough analysis or description would interest and benefit other engineers.

In such a case it is our normal course to seek out the engineer responsible for the fine new design, either by mail or by personal call, and to learn what information is available for publication without jeopardizing the interests of his company. "Dear Mr. Editor:" the engineer will reply, "Many thanks for your kind words about our new gadget. I will be only to glad to show it to you and to explain everything to you personally, but it will have to be with the understanding that nothing whatever is to be published. Our company is funny about such things and I am sure that you will understand. . . ."

Or again, it is now and then possible to persuade the engineer that a published description of his new design will not only be of service to his fellow-subscribers but will be of genuine benefit to his company. With his cooperation we whip an article into shape. We feel sure that it is a good article, but we are aware of a lack. Again we write to the engineer, this time asking permission to use his name.

"Dear Mr. Editor:" he writes, "I appreciate all you say about the use of my name, but I must ask you not to print it in this instance. Our company is funny about such things and I am sure that you will understand. . . ."

FRANKLY we are beginning to understand. We used to think that certain engineers were a bit nervous about the tenure of their jobs, that they did not wish to be placed in situations that might arise to embarrass them with their companies. But we now have reason to believe that we were mistaken. These cases are but further instances of that same old professional modesty. The engineer knows only too well that the executives of his company want publicity for their associates and their products—free publicity without stint or measure—and the freer the better. And what publicity can be more ethical or more justified than that which goes with the description of a genuine accomplishment?

We may be speaking out-of-meeting, *ex cathedra* and out of turn, but we feel that these examples of professional modesty have nothing to do with true modesty. Such modesty represents a warped state of mind, a definite and destructive complex. Dr. Freud named one complex, "Narcissus"; we call the engineer's complex, in the same language of flowers, the shrinking violet complex.

When the recently elevated engineer meets in the board room with the hard-driving, uninhibited and publicity-minded fellow officers from the sales and advertising departments he will see that one reason for their higher salaries is the fact that their complexes run to crysanthemums, sun flowers and other sturdy self-advertisers.

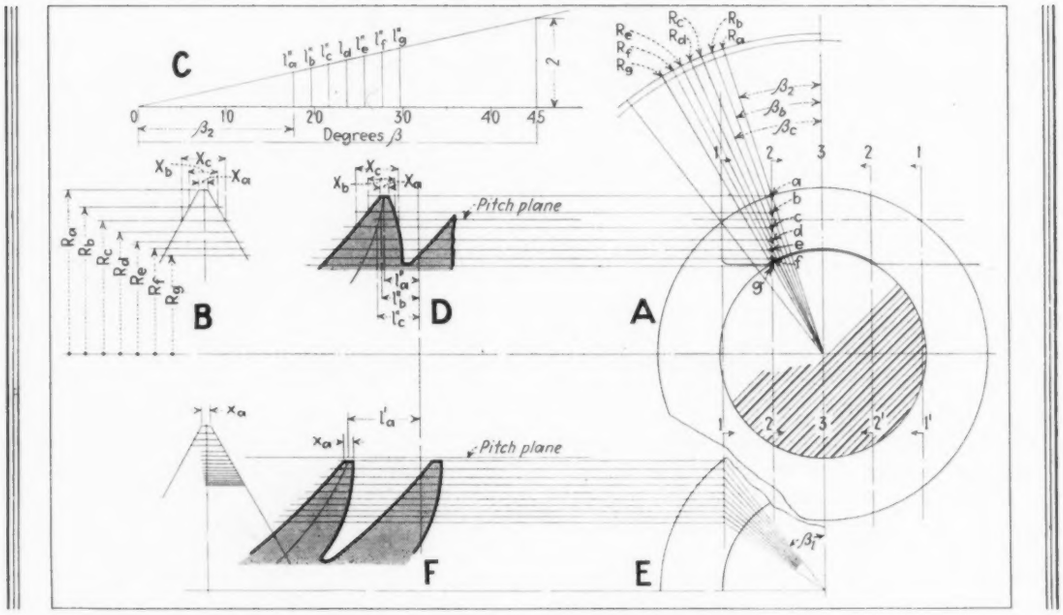


Fig. 1—Graphical method for determining the rack profiles of the worm in various planes parallel to the axial plane

Graphical Determination of Worm Gear Contact

WALTER M. POHL
Assistant Chief Engineer, Kearney & Trecker Corporation

In the design of worm gears, it is a simple matter to determine the contact between the worm and gear for the section in the central plane. But a complete analysis requires that this contact analysis must also be made for sections in other than the central or axial plane.

In projecting the various planes, such as those indicated by 1-1, 2-2, 3-3 in Fig. 1*A*, two fundamental characteristics of the cylindrical worm are made use of: The slope or shape of the thread profile is correspondingly the same in all planes of equal distance from the axis when viewed towards the axis of the worm, and any point on the thread profile has the same lead.

As an example, let us determine the worm and wormwheel sections and their corresponding action in the plane 1-1, Fig. 1*A*, which represents a view normal to the axis of the conventional screw type worm having straight side profiles in the plane containing the axis, as represented in Fig. 2. The proportions of the worm set, as illustrated, are as follows:

8 starts
1-in. linear pitch
O.D. of hob 3.700
Root diameter of worm 2.300
30-deg. pressure angle
Left-hand helix

It is imperative that the clearance must be added to the worm diameter in order that the true shape of the worm profile as generated by the hob will be represented. In laying out Fig. 1*A* to a convenient scale of enlargement, the various planes to be determined are marked as 1-1, 2-2, 3-3, etc., so that planes at equal distance from the center bear the same notation. This notation is founded on the fact that planes of equal distance from the center plane have identical profiles when viewed towards the axis. It is important in contact determinations as dealt with later that in constructing the profiles as at *D* and *F*, Fig. 1, the vertical or Y-axis passes through either the center of the tooth or the center of the tooth space of the profile for the section in the vertical plane containing the worm axis.

A radial line R_a is drawn to the point of intersection of the plane to be investigated, as 2-2, and the outside diameter of the hob. From the line R_a a sufficient number of arcs are drawn to cover the active portion of this plane, identifying their radii to the point of intersection with the plane as R_a, R_b, R_c, etc., so that a is the point of intersection of R_a, and b is the point of intersection of R_b, and similarly for other points.

Fig. 1*B*, which represents the tooth form in the center plane, and duplicated for clarity in explanation, is

Fig. 2—In the plane of the axis, the conventional worm profile has straight lines

drawn next with the corresponding true radii marked by horizontal lines with the tooth thicknesses X_a, X_b, X_c, determined at the corresponding radii.

The lead triangle as shown in Fig. 1*C* is now drawn with any convenient base of measurements in degrees, but maintaining the same scale of enlargement for the vertical ordinates as used in the drawing. The lead of the worm being 8-in., the triangle should read 2-in. to the scale of enlargement at 45 degrees.

Being a left-hand helix, the lead advance for β_2, β_b, β_c, is obviously to the left along the axis. Projecting lines from the points *a, b, c, d,* to the profile, the lead advance for the various angles is completed to the left of the centerline in Fig. 1*D*, as l''_a, l''_b, l''_c, etc., establishing the distorted centerline of the tooth. The tooth thicknesses determined in Fig. 1*B* are then plotted as X_a, X_b, X_c, in Fig. 1*D*, finally forming the ordinates of the true profile.

The same method of procedure is followed in plane 1-1 with its initial lead advance angle from the center plane designated as β_1, as shown in Fig. 1*E* and its initial linear lead advance l'_a in Fig. 1*F*. Occasionally it is found convenient in laying out the lead triangle, to establish various base lines as shown in the construction line in Fig. 1*C*, with the ordinates between the base lines equal to the lead advance β_1 to B_2, etc., thereby eliminating unnecessarily large triangles.

Having the rack profiles, the next step is to determine the line of action for its particular plane. Fig. 3 illustrates a method whereby no redrawing or tracing for the several intervals of profile movement is required. From the centerline are stepped off an equal number of divisions along the pitch plane. From these various points normals to the profile are drawn, designating the points of intersection, as *m, n, o, p, q, r,* and *s*. Since for true conjugate action the normal line must pass through the pitch point, the corresponding points of intersection are projected to the left or right equal to the distances of the pitch plane intervals from the pitch point, thus forming the ordinates of the line of action.

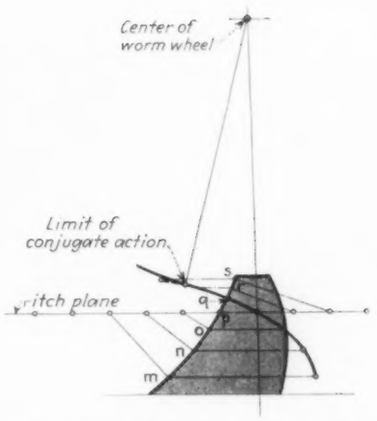


Fig. 3 — Method for graphically determining the line of action for several intervals of profile movement

The working limit of the line of action, or the so-called limit of conjugate action, is reached at a point of tangency of a circle about the worm gear axis with the line of action. Any protrusion of the worm profile beyond this point results in interference and undercutting of the worm gear.

In plotting the ordinates of the wheel profile, the conventional method of determining conjugate surfaces is used. Referring to Fig. 4*C*, equal divisions are plotted along the pitch plane of the worm as a_1, a_2, a_3, and similar mating points along the pitch cylinder of the gear as b_1, b_2, b_3. From the points a_1, a_2, a_3, etc., lines normal to the worm profile are drawn and their intersections designated as J_1, J_2, J_3. Projecting these intersectional points to the line of action, corresponding to the particular profile, the points k_1, k_2, k_3, are located through which arcs struck from the center of the worm gear are drawn.

Scribing the distances *a-j* from the

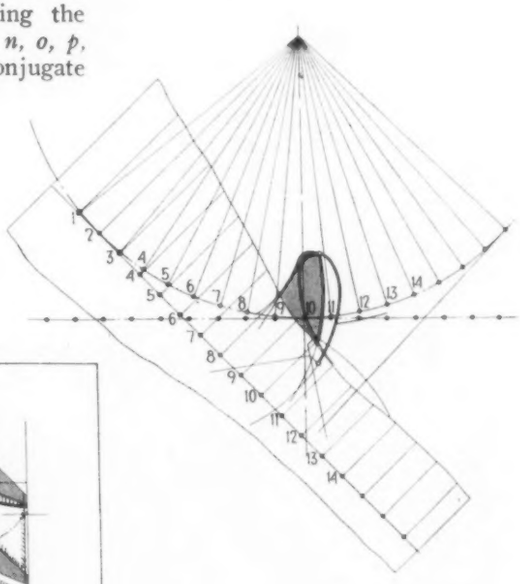


Fig. 5—(Above) The undercut portion of the profile can be determined by rolling a tracing of the interfering tip around the pitch cylinder

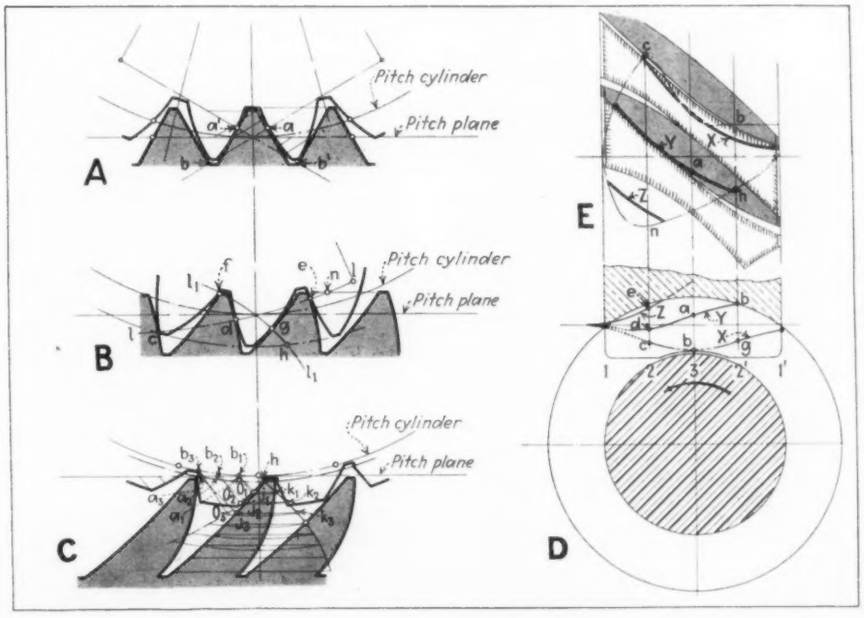


Fig. 4—(Left) In plotting the ordinates of the wheel profile, the conventional method of determining conjugate surfaces is used. At *D* are shown the contact lines plotted from points of contact in *A, B,* and *C*. The boundary of contact is shown at *E* where are also shown the actual tooth sections parallel to the worm axis

points b, the intersections with the arcs drawn through k finally form the ordinates of the worm gear q_1, q_2, q_3, etc. When existent, the undercut portion of the profile may be determined rapidly by rolling a tracing of the interfering tip around the pitch cylinder, as in Fig. 5, and piercing the successive stages of the tip with a needle point. Lines normal to the pitch plane are drawn through the interval points on the tracing of the worm and brought radial at the corresponding interval point on the pitch cylinder. This method may also be used to check the active profiles of the gear.

Fig. 4*D* shows the contact lines *X-Y-Z* as plotted from the points of contact in Figs. 4*A*, 4*B*, and 4*C*. The direction of rotation is assumed as shown by the arrow. It is obvious that for opposite direction of rotation the lines are viewed 180 degrees in the plane of the paper.

The zone of contact, or in other words the boundary of contact, is shown in Fig. 4*E*. The ordinates of this boundary for any particular plane, as say Fig. 4*B*, are the distances from the plane of the worm gear axis to the intersection of the tip of the worm profile with the line of action *l-l* for limits *n*, and the intersection of the extremity of the wheel tooth with the line of action *l-l* for limit *c*. For the plane 2′ the intersection of the tooth extremities with the line of action *l-l*, in Fig. 4*B*, form the limits *f-h*. It should be understood that when the limit of action is reached by a condition as in Fig. 3 within the scope of the profile, this then becomes the controlling factor for the length of contact. By projecting the contact points from the plane of the worm gear axis in Figs. 4*A*, 4*B*, and 4*C* contact lines *X-Y-Z* are plotted in the zone. Needless to say, the contact points must be consistent with the same tooth in all planes for a given contact line.

For a study of curvature ratio of contact, it is often desirable to plot the actual tooth sections parallel to the worm axis, as shown in Fig. 4*E* for illustrating a section through the pitch plane of the worm, which again is projected from Figs. 4*A*, 4*B* and 4*C*. Fig. 6A shows the contact lines as viewed when plotted on the solid worm.

It will be found by some, who prefer mathematical analysis, that the above method will aid in a clearer understanding of the subject for either partial mathematical solution or complete solution as treated in *Product Engineering*, Vol. I, Nos. 1, 2, 3 and 4, by Prof. Earle Buckingham.

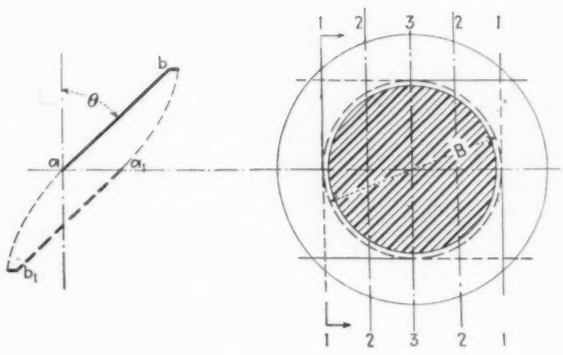


Fig. 6—In all involute helicoids the generating sides of the profile are straight in all planes tangent to the base cylinder and inclined at an angle equal to the helix angle at the base cylinder

Involute and Convolute Helicoids

The derivation of the rack sections of the involute helicoids is based on an important feature of this type of thread form.

In all involute helicoids the generating sides of the profile as *a*, *b*, and a_1, b_1, Fig. 6 are straight in all planes tangent to the base cylinder *B* and inclined to an angle equal to the helix angle at the base cylinder; or if a bit of mathematics may be injected under this heading, $\tan \theta = Lead/\pi B$.

Properly to proportion the tip thickness of the thread, the curved portion in this plane is next determined. Advancing the generating side *g*, *o*, Fig. 7*A*, in a clockwise direction, the angle of advance is noted as β_1, β_2, etc., and the intersection of the advancing tangents with the plane 1-1 as *a*, *b*, *c*, *d*, etc. From the straight side g-o_1, in Fig. 7*B*, draw reference line *lg* inclined to the angle θ from the vertical and intersecting the side at the plane of the axis. Lines parallel to the axis are next drawn through the intersection points *a*, *b*, *c*, etc., to the reference line. From the lead triangle Fig. 8, the lead advance for successive points is measured and plotted to the right of the reference line, forming the ordinates of the curved profiles. The relation of tip thickness and base line thickness, unless determined previously by some definite standard, may then be determined by the location of the upper straight line portion *g*, *o*, along the axis.

In determining the rack section of plane 2-2, the initial lead advance from plane 1-1 to plane 2-2 is plotted in Fig. 7*D* measured from the corre-

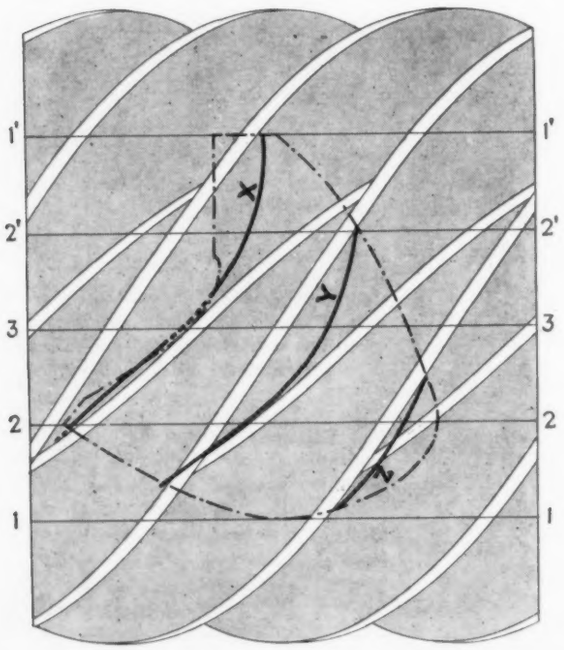


Fig. 6A — Contact lines in their ultimate form as they appear when looking down on the surface of the solid worm

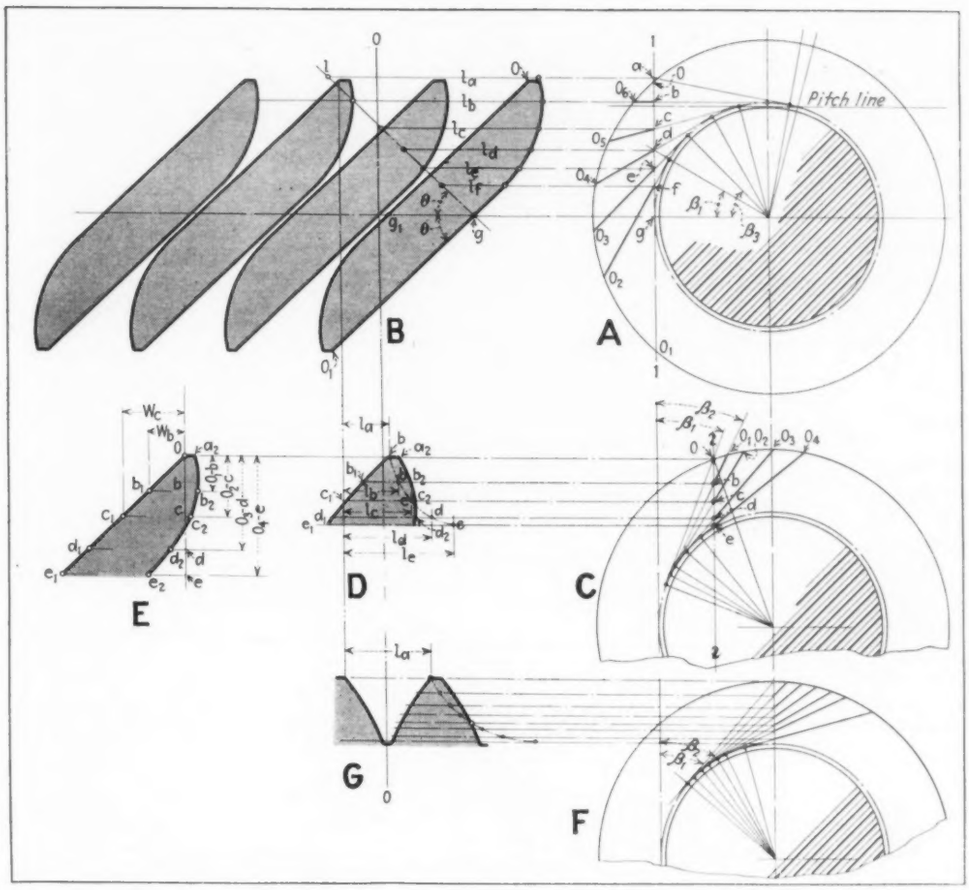
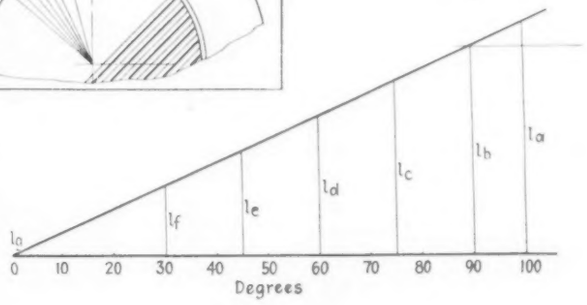


Fig. 7—Graphical method for determining the rack sections of the involute helicoids. The lines of action and conjudgate gear profiles are determined in the same manner as in the screw helicoid, and therefore are not shown

Fig. 8 — From the lead triangle the lead advance for the successive points is measured and plotted to the right of the reference line, forming the ordinates of the curved profile



sponding tip line projecting down from Fig. *7B*. With the initial lead advance for β_1, and the successive increments for β_2, etc., plotted along their respective intersection lines of the tangents and plane 2-2, the points of the distorted tooth centerline are established. Next the tangential distances o_1-b_1; o_2-c; o_3-d; etc., from the periphery of the worm to the plane under investigation are measured downward from the top of the tooth in Fig. *7E*, an identical section as *7B*, and lines parallel to the axis drawn indicating tooth thickness for these intersectional points as b_1-b_2; c_1-c_2; and their location from the centerline as bb_1, bb_2, etc. Plotting the horizontal ordinates bb_1, and bb_2, from the distorted centerline in Fig. *7D* along the intersectional lines outlines the actual profile.

For the center plane 3-3 the same procedure as for plane 2-2 is used, and as in Fig. *7F* the distorted centerline and profile is resolved.

The lines of action and conjugate gear profiles are determined in the same manner as in the screw helicoid and therefore not shown. As the plane of generatrix is at a distance from the plane containing the axis, the wormwheel center *O-O* must be drawn after the completion of Fig. *7F*. It is imperative in this, or any type of thread form, that the worm gear centerline must pass through either the center of space or tooth in the plane containing the axis in order that correct contact conditions are depicted for planes equal distance from the center plane. Unless this is done, all planes to be determined must be drawn to the full extent, entailing needless effort and time. The contact lines for the involute helicoid analyzed in this article are shown in Fig. 9 but are by no means characteristic for all involute helicoid proportions.

The convolute helicoid, or thread form having straight sides in planes tangent to the base cylinder, but at a different angle than helix angle at the base cylinder, is treated in the same manner as above, with reference to line *lg*, Fig. *7B* at an equal angle from the axis as the straight portion go_1.

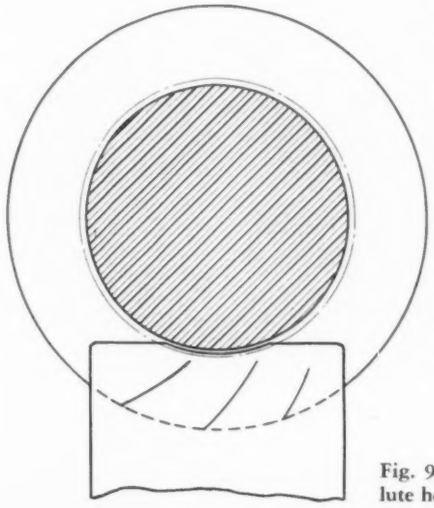


Fig. 9—Contact lines for the involute helicoid analyzed in this article

PRODUCT DEVELOPMENTS

Hydraulic Throttle Control—Hermetically Sealed Transformers—Recent Applications of Parts and Units—Built-In Frequency Changers—"Unicams" for B & S Automatic Screw Machines—Splash-Proof Telephone Dial—Stainless in Two Unconventional Uses

▸ Hydraulic Throttle Control

It is quite a problem to design an efficient throttle control for airplanes, due to the large number of push-rods, bell-cranks, springs, joints and bearings required. The slightest wear is amplified, making it difficult to avoid sloppiness and consequent lack of precision in throttle adjustment. When one attempts to synchronize the three engines of a tri-motor plane, troubles from this source become even more apparent.

In the Wynn hydraulic throttle control, as shown in the accompanying drawing, the throttle lever in the cockpit is connected by suitable linkages to a pair of plungers operating in two small parallel cylinders. As indicated, the lever arms are so arranged that as one plunger moves inward in its cylinder, the other moves outward. The butterfly valve at the carburetor is linked up through a yoke to a corresponding pair of cylinders. Copper tubing connects each carburetor cylinder with one of the throttle control cylinders. When each cylinder and tube is completely filled with a non-compressible, non-freezing fluid, it is obvious that an angular displacement of the throttle lever will produce a corresponding angular displacement of the butterfly valve. Each cylinder is fitted with a bleeder valve of the needle type *A* to permit filling of each leg of the system completely without inclusion of air bubbles. A supply pump is used to fill the system on the ground. Rubber cup packings prevent leakage in the cylinders.

The two hydraulic elements of the system are completely balanced at all times, and are under no pressure to cause leakage. If necessary, the system will still continue to function with one of the hydraulic connections completely broken. Sufficient vacuum can be attained in the operating cylinder, so that a movement of the lever in either direction can be obtained under an emergency condition of this sort. Calculations indicate that the system may be installed on an average three-engine airplane at a total weight of about the same as the mechanical system which it replaces.

▸ Hermetically Sealed Transformers

It is reported that hermetically sealed transformers are appearing in British practice. These eliminate contact between insulating oil and at-

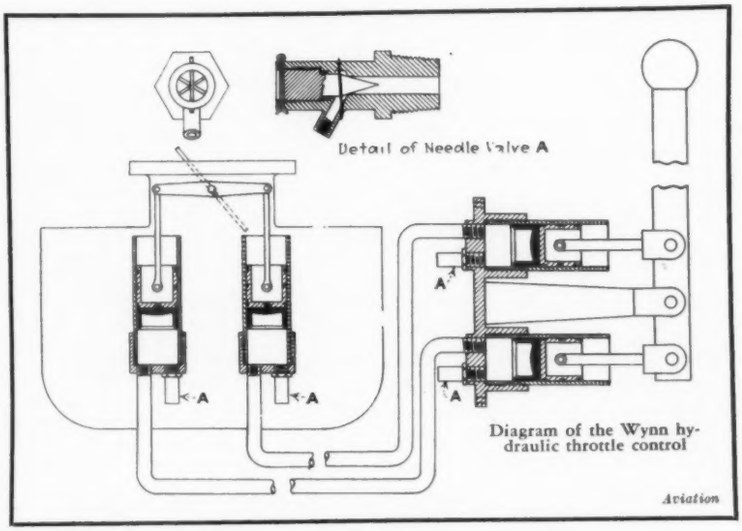
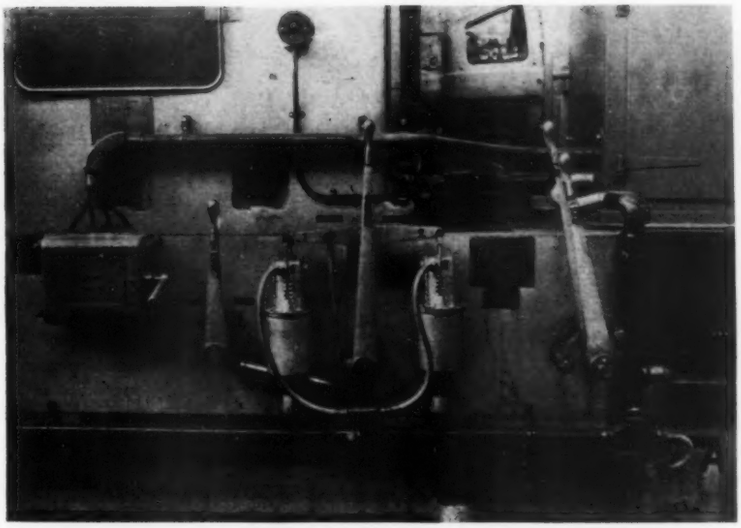


Diagram of the Wynn hydraulic throttle control

Aviation

Probably the largest electric shovel ever built. Dipper capacity 18 cu.yd.; weight 1,100 tons; height 80 ft.; 3,000 hp. in General Electric Ward Leonard type electrical machinery including 800 hp. synchronous motor. Use of counterweight on hoist motion is recent development which reduces peak demands on public service. Built by Marion Steam Shovel Co. for Clemens Coal Co.



One operator can control this English boring machine in all its motions by means of pendant-type push-button stations which operate Thrustors. The operation of one or other of the Thrustors moves the lever in a clockwise or counter-clockwise direction as the case may be to energize either the forward or reverse feed

Aviation

The 38 per cent negative setting of the top wing of the Beechcraft airplane is a marked departure in American commercial practice. Advantages include better pilot visibility forward, better location of front spar of top wing with relation to fuselage and landing gear, and improvement in air flow over top wing at angles near the stall

mosphere, wastage of oil, maintenance of the dielectric and breathing of the apparatus. The engineers of the British Electric Transformer Company have designed a flat round oil container attached at its center to the transformer tank lid, so that its interior communicates with the main tanks and is completely filled with oil. The container is constructed of two corrugated copper disks, joined together on their outside edges by a solid band 1 in. wide and left free to expand outward from each other in the center. The plane of the disks is always horizontal, so that the fluid pressure inside is equal at any point of the periphery. Three lugs on the periphery of the disks are drilled to slide freely over three steel standards fitted to the tank lid, which act as guides to the disk as it rises and falls according to the temperature.

The diameter of the disks is so arranged that for a 50 deg. C. rise in tank temperature the back pressure from the flexible diaphragms when expanded does not in any case exceed 2 lb. per sq.in. The temperature rise in the expansion device is never so great as in the transformer tank itself, the figure usually reached being about two-thirds of the tank temperature owing to the greater proportion of cooling surface exposed by the disks.

Special arrangements have been made by the company for filling and sealing these transformers, so that at ordinary temperatures the tank is under a slight vacuum of about ¼ lb. per sq.in. This is found to be an advantage, since it reduces the back pressure reached at higher temperatures by a corresponding amount.

▸ Recent Applications of Parts and Units

A photoelectric cell has been adopted by the engineers of Rice, Barton & Fales for reversing the action of their "dye jigg" for textile mills. A beam of light is projected across the underside of the beam roll, and as the roll of cloth gradually decreases in diameter, as it is paid out, it finally gets small enough to let the ray of light shine across the machine. The ray then strikes the photo-cell which operates a relay, reversing the motion of the machine and thus keeping the cloth material to be dyed moving continuously back and forth through the dye liquid.

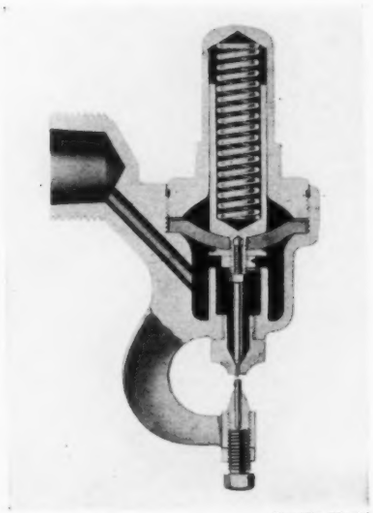
The new 8-head rotary bottle crowner exhibited by The Liquid Carbonic Corporation is designed with a Reeves variable-speed transmission integral in the base of the machine. Since any speed from 45 to 150 bottles per minute is thus made possible, it has been remarked that the crowner is ready for any turn in the repeal wind.

Another application of the self-starting synchronous motor has been revealed in the "Frostoff Defroster," an automatic device for defrosting electrical refrigerators recently marketed by Waterbury Clock Company. Once every 24 hours the motor operates a switch which interrupts current flow into the refrigerator long enough to permit defrosting.

▸ Built-In Frequency Changers

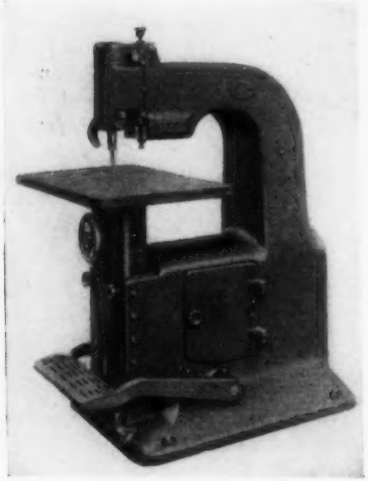
Frequency changers, for changing 60-cycle current to a higher frequency, thereby increasing the speed of the synchronous motor proportionally, are used to a considerable extent. Woodworking machinery, with the high-speed spindles driven directly by the motor, are a common example. Thus, a 60-cycle, 3,600-r.p.m. motor will give a speed of 10,800 r.p.m. when the frequency is stepped up to 180 cycles.

It has been quite common to let



Textile World

Self-cleaning humidifier nozzle designed by the American Moistening Co. Water pressure acts upon rubber diaphragm to reverse normal position, thereby lifting cleaning pin from orifice channel. When water pressure is relieved, diaphragm assumes its natural position, thus forcing cleaning pin into orifice channel

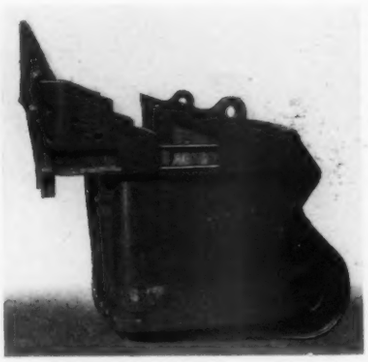


With built-in frequency changer in the base of the machine, this router manufactured by the Yates-American Machine Company is a completely self-contained unit. Note the design of the motor

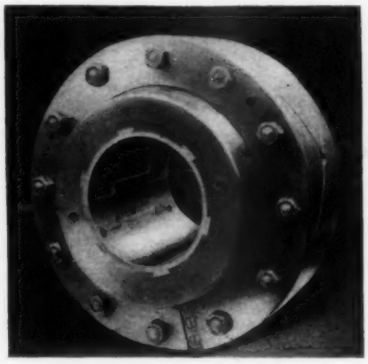
the man who installs the machine cast about for a suitable location for the frequency changer. Sometimes it will be mounted on the floor behind the machine, sometimes set on an overhead shelf, sometimes bracketed to the wall or even suspended from the ceiling. Now, however, in one of the latest designs of routers, the frequency changer is made an integral part of the machine.

This new high-speed electric router, a product of the Yates-American Machine Company, marks an innovation. The frequency changer is mounted within a frame, protected from dust and dirt, and rests on full-floating rubber supports to absorb vibration. At the mere touch of a button either 10,800 or 18,000 r.p.m. on the spindle is obtained.

Another feature of the design of this router is that the motor and spindle are built into one compact unit which moves up and down on the vertical gibbed ways, this direct motor drive giving assurance that there will be no reduction of speed and power under load as might be caused by a belt drive. The motor frame is of special design with a disk-type brake built in at the upper end of the housing, and a blower for cooling the motor and bearings and blowing away chips built-in in the lower portion of the housing.



Amsco one-piece renewable lip power shovel dipper is so designed that the digging front may be renewed in the field without the employment of riveted connections. In addition the operator may quickly apply a different type lip and base section when encountering a new earth or rock condition



One of the self-aligning assembled bearings for a Sperry gyroscopic ship stabilizer. Details of the design of these bearings were given in the January *Product Engineering*

▸ "Unicams" for B & S Automatic Screw Machines

When one analyzes a large number of Brown & Sharpe lead cams, it is found that the same lobes recur many times. The engineers of the Fischer Special Mfg. Co. recognized this fact and developed separate cam lobes of standard lengths and rises. By assembling these individual lobes on a spool, a lead cam of the required number of lobes can be built up which will do the work of a solid cam.

To use this built-up cam, a lead cam lever with a wide roller and several other parts are substituted for the ones which come as original equipment on the machine, after which either "Unicams" or solid cams can be used. It is claimed that where Unicams are adopted instead of solid cams, savings can be made not only in the cost of cams but also in the time saved by being able to get new lobes from stock. The user soon accumulates a number of lobes, many of which can be used on subsequent jobs.

Different lobes are made with rises and falls for fast and slow machine times; any number of lobes up to six can be built up. The cams are made with standard rises up to 3 in. in height, with a variation between rises of 0.020 in. The reason for using 0.020 steps in the rises is that this variation allows some clearance on nearly all fractional sizes. The lengths in hundredths are in multiples of 3 up to 15, and in multiples of 5 from there to 80. The hole in the Unicams is broached with 100 teeth, and the spool has the same number of teeth so that the lobes are adjustable around the cam circle by single hundredths.

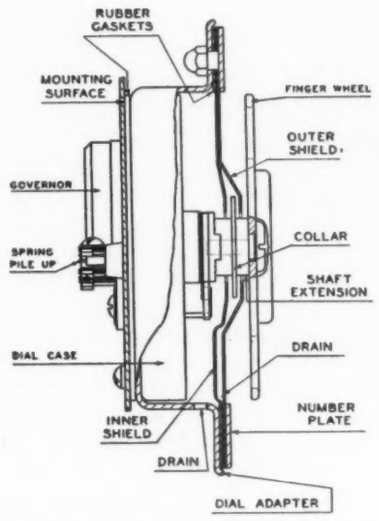
▸ Splash-Proof Telephone Dial

Although the deck telephones used in Navy service are mounted in water-tight boxes, the cover of the box must be open in use, exposing the dial to spray or rain. Thus, the engineers of the Bell Telephone Laboratories, Inc., modified the standard dial design to make it "splash-proof." In this it was desirable from the standpoints of both manufacture and cost to make a minimum number of changes, and obviously necessary to make them so as in no way to interfere with the functioning of the dial. Thus, while a stuffing box on the

dial shaft would be effective in excluding water, it would introduce variable friction unfavorable to the maintenance of constant dialing speed. For this reason it was necessary to build shields and baffles around the dial to deflect water striking the outside.

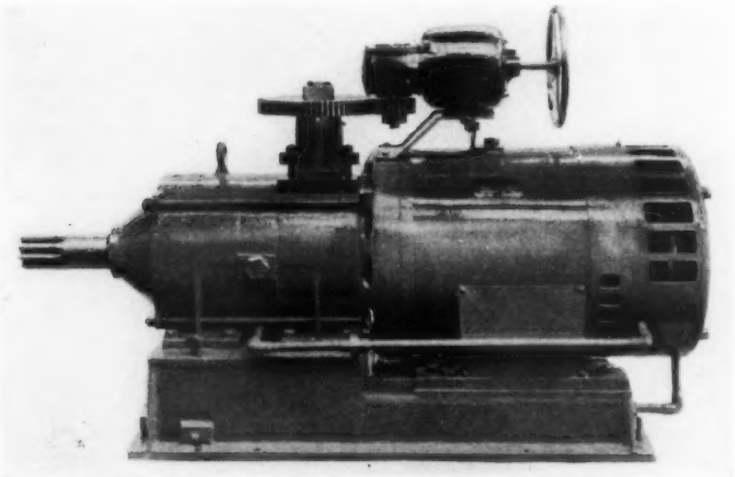
As shown in the illustration, a dial adapter developed previously for use on coin collectors at public telephones was found well suited as the base of the splash-proofing construction and the mounting for the number plate. The standard dial, from which the finger-wheel, finger-stop and number plate have been removed, is placed in the cup of the dial adapter. A rubber gasket and a thin plate, the "inner shield," are then mounted on the flange of the cup. A short "shaft extension" is placed on the dial shaft; the outer shield and number plate are assembled on the flange. A collar machined on the shaft extension is located halfway between the inner and outer shields and overlaps the edges of the hole in each. The dial finger wheel is mounted on the end of the shaft extension, and a special finger stop is fastened to the number plate.

In order to prevent water from entering the back of the dial, a rubber gasket is interposed between the dial adapter and the subset on which it is mounted. If water is thrown against this dial, which is always mounted in a vertical plane, most of it will be deflected by the outer shield and will run off across the number plate. Some water may pass through the opening between the hole in the shield and the dial shaft, and this will be deflected by the collar on the shaft and will drain through a radial duct formed in the inner shield and ending in a hole in the outer shield. Finally, a small hole, drilled in the bottom of the adapter cup, permits the drainage of any small amount of water which might possibly enter the interior of the dial. Thus, while the modified dial is not water-tight, the shields make it as nearly splash-proof as practicable, and it is giving satisfactory service to the Navy.



The designers of this "splash-proof" telephone dial for Navy use avoided the use of a stuffing box which would have interfered with the functioning of the dial

A taper wedge and a Woodruff key secure the pin in Hadfields Ltd. (Sheffield, England), dredger buckets and prevent the movement of the pin when the buckets are in service



25 hp. Hele-Shaw electro-hydraulic transmission with remote control furnished by American Fluid Motors Co. Delivers any speed from 0 to 870 r.p.m. Reversible instantaneously without gears

▸ Stainless in Two Unconventional Uses

Word comes from England that cast stainless steel is the material used in the largest single-runner Pelton waterwheels ever built in that country. The units, four in number, are designed to develop 17,000 hp. each and are to be installed on the Uhl River, Punjab, India. Each wheel consists of a solid steel disk 6 ft. 6¾ in. in diameter, to which are bolted 22 buckets. These buckets are cast in pairs with lugs that embrace the disk, and each pair is held by three bolts. The reason for making the buckets in pairs and sharing the lugs is that if each bucket had its own lugs each lug would require two bolts. There would then not be sufficient room around the circumference of the disk to accommodate them.

Stainless steel may also be said to have found unusual employment in the blade of a single wing 1 kw. wind power electric generating unit recently placed on the market by the Wind Turbine Company. This blade is of air-foil section, hollow, and built up of stainless steel fabricated by the Budd "Shotweld" process. At any wind speed above about 15 m.p.h. the mill speed is constant, the blade automatically feathering with increased wind velocity. Feathering is accomplished by balancing wind pressure and weight against an adjustable spring.

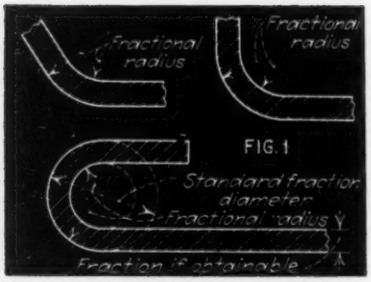


Fig. 1—If the outer radius is in fractions, standard fractional size tools can be used to obtain the die shape. If the stock thickness is also a fraction, standard fractional milling cutters can be used

WHY WELL-ROUNDED DESIGNS MAKE ECONOMICAL STAMPINGS

J. K. OLSEN
Chief Draftsman, Stewart-Warner Corporation

This fifth article takes up the question of corner radii in stampings and shows how considerable savings in material can be made

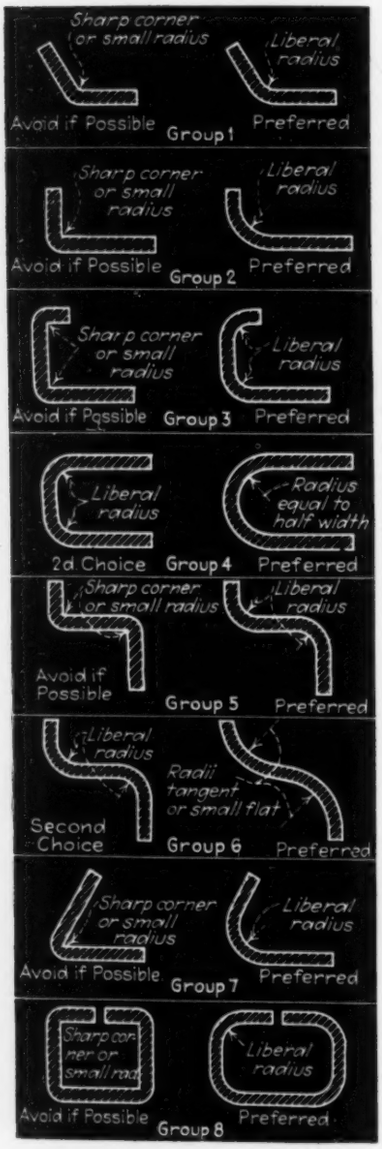


Fig. 2—A few typical examples of the application of radii in bends and forms, indicating constructions to be avoided and those that are preferred

When designing stamped parts with bending or forming operations it should be the rule to use the largest possible radius, within reason, at all corners and bends. Adherence to this rule will almost always prove economical in raw material, cost of dies and die upkeep, and will also make it possible to eliminate a large amount of scrapped parts. Often where a part with sharp bends is made it requires softer and more expensive material. Large radii in the design permit a wider selection of suitable raw materials. They allow the use of a harder and stronger temper stock, save in the original cost and generally make a part that is stronger.

Standard Sizes for Radii

Another item in the production of stamped parts which is overlooked in most instances is the adoption of standard sizes for the radii in formed or bent parts. It is of tremendous importance in manufacturing to keep the tool expense as low as possible. The designer should not use special decimal-size radii unless there is a good reason, but should adhere to common or standard sizes, since drills, reamers and milling cutters in standard sizes can be obtained more readily and at less cost. When special sizes are used it usually requires that new tools be made or purchased for use in making the punches and dies. A good rule to follow is: "Always favor common-fraction sizes." This applies to almost every item of the design.

In dimensioning radii for a bent or formed part it is a simple matter to adopt fractional sizes and in this manner make use of standard tools on hand. The standard fraction size can be specified where it is most advantageous, that is, it may be on the outside of the bend in some instances and on the inside of the bend in others.

The use of fraction sizes, specified either as a common fraction or as a decimal, is preferred not only for radii but for all dimensions. There is at least one main advantage, and that is in the checking and inspecting of the part. It is general practice in manufacturing to hold decimal dimen-

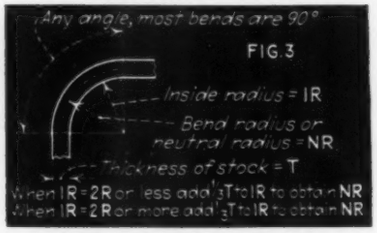

Any angle, most bends are 90°
FIG. 3
Inside radius = IR
Bend radius or neutral radius = NR
Thickness of stock = T
When IR = 2R or less add ½T to IR to obtain NR
When IR = 2R or more add ½T to IR to obtain NR

Fig. 3—The method shown here for figuring the length of stock is close enough for ordinary estimating

sions closer than fractional dimensions.

In the bends and the hook illustrated in Fig. 1, if the outer radius of the bend is given in a fraction it enables the use of standard fraction-size tools to obtain the die shape. If, in addition, the thickness of the stock is also a fraction, a standard fraction-size milling cutter can be used. In this connection it should be pointed out that, in most instances, at the start of the design the designer can arrange to use fraction sizes, and usually it would be more practical to use only common fractions for all sizes and dimensions.

The illustrations in Fig. 2 are a few common applications of radii in bends and forms, indicating constructions to be avoided and those to be preferred. The designs shown cover the conditions to be found in most drawn parts. The old rule, that for a given length of line a circle bounds most area, also can be applied in the forming of stampings. The circular or well-rounded design should always be preferred. In all of the eight groups shown, the second figure in each group should be favored since that design uses less material, giving a more economical material layout, and the parts will be easier to make.

Liberal Radii Save Material

In bending and forming parts similar to the ones shown, it is better practice to give a small radius rather than to have a sharp corner, while where possible the liberal radius or the complete half round is most preferable. Of course this depends on conditions. It should be remembered that a part with a sharp corner or a small radius most likely would have to be made from annealed and more expensive material, whereas the same part having liberal radii could be made from a harder material.

The length of the arc on neutral radius is the length of the stock required to make the bend specified and may be figured for any angle. The other lengths of the developed blanks are obtained by taking the sum of the straight lengths and adding this to the length of stock required for the bends or forms. A similar method may be applied when estimating drawn parts.

There are many different methods in use for figuring length of stock required for parts that are bent or formed, but the method for figuring neutral radius shown in Fig. 3 is a practical method that has proven itself satisfactory in extensive use, and is close enough for ordinary estimating. It should, however, be noted that where the stock required in bends, forms or draws is of importance, the "cut-and-try" method is the only sure way.

The designs illustrated in Fig. 4

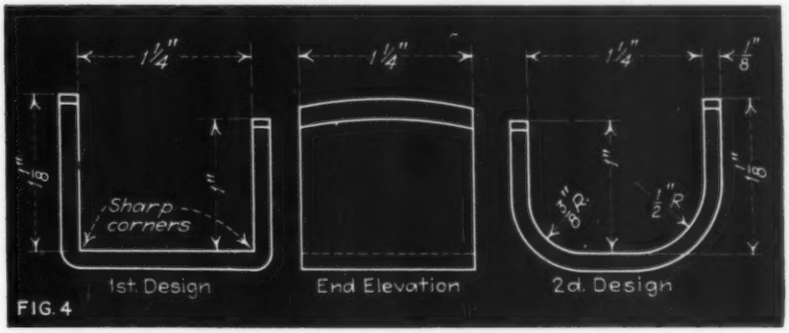


Fig. 4—Two designs, exactly alike except for rounding the corners. The length of stock required is shown in the figures below

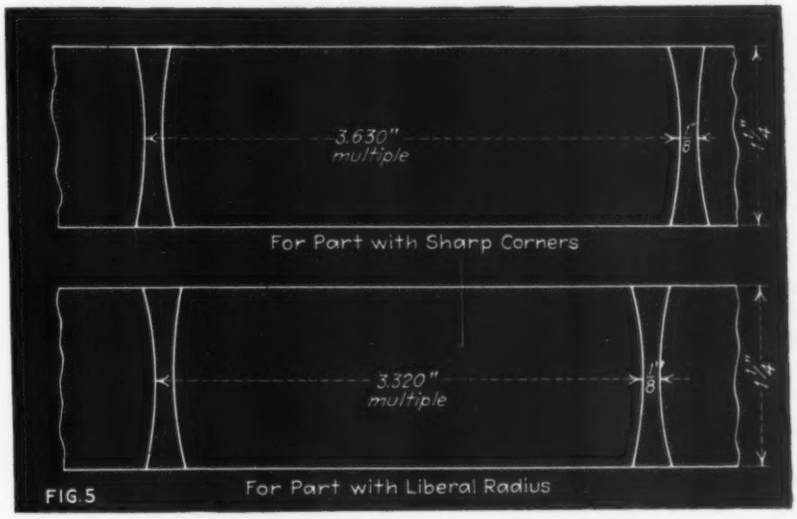


Fig. 5—Here is shown the saving in material when the corners are rounded as in the second of the designs shown in Fig. 4

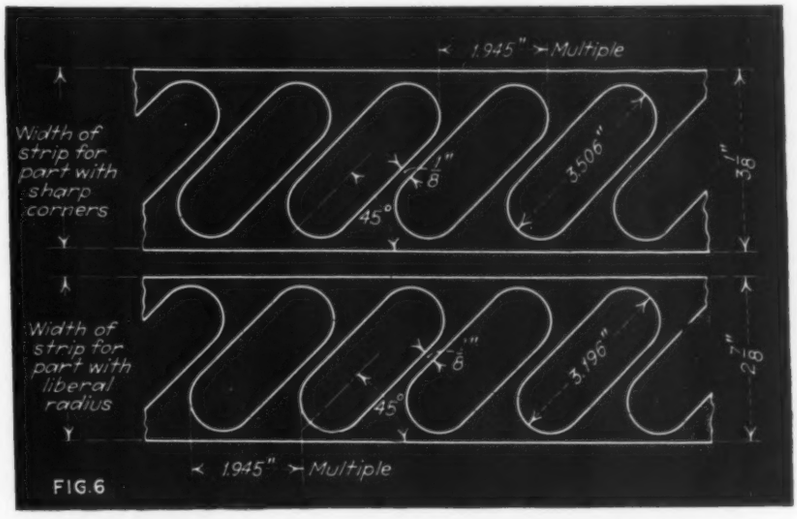


Fig. 6—Rounding the ends to semi-circular shape and placing the blanks at an angle, results in further savings of material

are selected as examples to show how the rounding of the corners saves material. In the first design with sharp corners the developed length L is obtained by adding the straight lengths plus the length required to make the two bends, which equals:

$$L = 1.125 + 1.250 + 1.000 + 0.065 + 0.065 = 3.505$$

In the second design, with liberal radii in the bends, the developed length L is obtained by adding the length of stock required to make the bends plus the straight portions which are obtained by subtracting the radius in each corner from the two sides and the bottom.

$$L = 0.687 + 0.883 + 0.625 + 0.375 + 0.625 = 3.195$$

The comparative material layouts for these simple designs when made from $\frac{1}{8}$ x $1\frac{1}{4}$ bar steel would be as shown in Fig. 5. As can be seen by the two material layouts, the part with the sharp corners requires a longer piece of raw material as compared with part having liberal radii in the corners. The first design with sharp corners would require multiples 3.630-in. long for each part and would use 169 lb. of stock per 1,000 pieces including a 5 per cent allowance for waste. The second design with a liberal radius in the bends would require multiples 3.320-in. long for each part and would use 154 lb. of stock per 1,000 pieces including a 5 per cent allowance for waste. By using this design a saving of 9 per cent in raw material would be gained. These same parts, except with ends rounded, when stamped out of strip, and staggered at the same angle, would have material layouts as shown in Fig. 6. In the first design, the part with sharp corner bends would require a $3\frac{1}{8}$-in. wide strip and 1.945-in. multiples. This would use 226 lb. of stock per 1,000 pieces including a 5 per cent allowance for waste. The second design with liberal radius in the bends would require a $2\frac{7}{8}$-in. width of strip and 1.945 multiples and would use 208 lb. of stock per 1,000 pieces including a 5 per cent allowance for waste. With this design there would be an 8 per cent saving in material.

It frequently happens that a lug or ear must be sheared out of the side or base of the part. In such case the designer must figure out the space required so as to be able to incorporate this feature in the design. In Fig. 7 the length of A plus allowance for bends plus length C must equal the length of the opening L from which the lug or ear is sheared.

Rounded Profiles

A well-rounded design is a feature worth bearing in mind when designing stamped parts. The use of rounded features are advantageous in other respects besides in bends, forms or draws, namely, to make use of radii on the shape of the part itself. Raw material savings and minimum die cost can often be obtained by giving this feature the proper consideration. In other words it can almost always be said that the merits of a stamping designer can be seen by how judiciously he uses his compass.

The shaping of parts depends to a certain extent on the designer's likes or dislikes of shapes or forms. Some designers will go to the extreme of making ornamental shapes regardless of the practicability of the part. Undoubtedly the desired result, in addition to having a more practical part, could be had by the wise use of well-rounded corners. As an illustration, the four parts shown in Fig. 8 can be taken as typical examples showing how a small change in shape will save both on material and cost of the dies. These four common angle designs would in general serve the same purpose, but in quantity production each would differ in cost. The material layouts for the four angle-shaped parts would be as pictured in Fig. 9.

A comparison of the four material layouts shows the following differences:

A—This layout uses more material than any of the others when staggered in the conventional manner shown. It uses the widest strip.

B—The second layout with radial ends permits the use of a strip of less width than the first layout, and *A*, *B* and *C* have the same multiple.

C—The third layout with rounded ends makes use of a still narrower strip than *A* and *B*.

D—The fourth layout uses the same width as *C* but its shape permits a closer multiple.

These simple designs of angle-shaped parts show the importance of considering the use of well-rounded designs. It will be found that there are numerous opportunities when designing stamped parts to incorporate this factor of saving which also usually results in improving the parts.

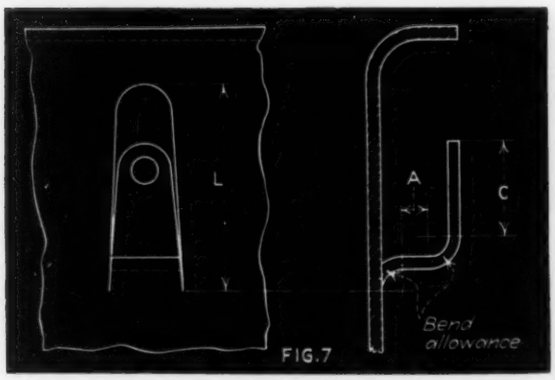


Fig. 7—The length *A* plus allowance for bends, plus *C*, must equal length of opening *L* from which the ear is sheared

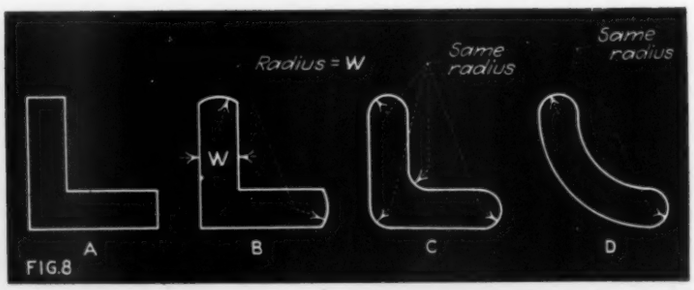


Fig. 8—These four designs show how a small change in shape affects costs

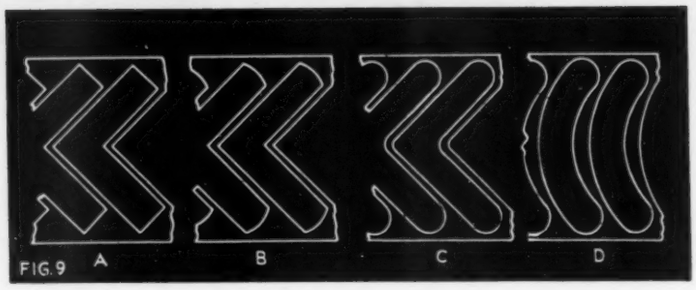
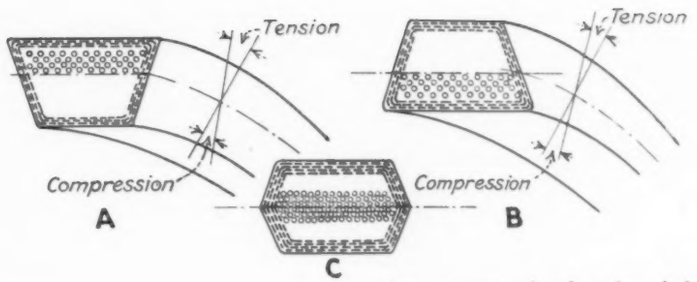


Fig. 9—Relative material layouts for the four angle-shaped parts shown in Fig. 8

CHARTS TABLES DATA for



For reverse bends the belt section shown at *C* must be used

Designing V-Belt Drives

ALBERT HACK

There are two distinct steps in the design of every V-belt drive. One is the proportioning of the pulleys, the selection of pulley diameters and the determination of the minimum center distance. The other is the calculation of the size and the number of belts required.

Proportions are important if a successful and economical V-belt drive is to be obtained. Ordinarily, the center distance should be equal to or slightly greater than the diameter of the large sheave. When the drive ratio is close to unity, there need be only a slight clearance between the faces of the pulleys, but as the ratio increases the center distance must be increased correspondingly in order to obtain a sufficient arc of contact between the belt and the small sheave.

For a given speed of rotation, the belt speed will be determined by the pulley diameter. If possible, the belt speed should be kept between 1,000 and 5,000 ft. per min. Although V-belts will operate satisfactorily above and below these speeds or limits, the cost usually becomes prohibitive. The most economical belt speed is between 3,500 and 4,000 ft. per min., both as to first cost and upkeep.

Belt manufacturers usually specify a minimum pulley diameter for a given size belt. Although it is the opinion of many engineers that smaller pulleys should not be used, special applications or conditions will sometimes make it profitable to disregard this rule. Economy of material, lack of space or other considerations might make it desirable to use pulleys smaller in diameter than the recommended minimum. But such designs must be studied with great caution, and a proportional loss in horsepower capacity must be allowed when making the calculations for such a drive.

After some years of extensive study and research work on V-belts and drives the writer devised the accompanying chart. The data on which this chart is based were taken on the performance of the layer-built type of V-belt. The variation of the life of layer-built belts in most cases is not more than 5 per cent, and in the construction of the chart a corresponding allowance was made to serve as a factor of safety. The chart also covers loss of arc of contact in both V-drives and V-flat drives. The table and auxiliary graph on the chart provide a correction factor for loss of horsepower resulting from small pulley diameter. This correction factor should be used whenever the pulley diameters are less than the minimum recommended by the belt manufacturers. The loss in horsepower capacity is caused by the excessive stress resulting from sharp bending of the belt while passing over the smaller pulley, change in the side angle and other factors. These are distinct losses and should not be confused with the decrease in horsepower capacity resulting from the lower belt speed when the pulley diameter is decreased.

Recommended Belt Sizes

Horsepower	Belt Cross-Section Recommended
Up to 2	A
2 to 5	A or B
5 to 10	B
10 to 25	B or C
25 to 50	C
50 to 100	C or D
100 to 150	D or E
Above 150	E

With reference to the table of recommended belt sizes, these are average figures for the layer-built type of belt. Other types of belt construction and the belts of certain manufacturers might call for some modifications of the size of belts to be used. However, this would not effect the subsequent calculations. Similarly, for different working loads per belt, the curves *A*, *B*, *C*, *D* and *E* can be reconstructed. The correction factors applied for undersize pulleys will not vary to any appreciable extent.

As an example of the use of the chart, assume a fan is to be driven by V-belts, the following data being given:

Maximum horsepower, 30.
Outside diameter of small pulley, 9 in.
Outside diameter of large pulley, 22 in.
Center distance, 21 in.
Belt velocity, 3,000 ft. per min.

For a 30-hp. drive, the proper size of belt is *C*, or a $\frac{7}{8}$x$\frac{5}{8}$ in. belt, as found from the accompanying table.

To design this drive by the use of the chart, starting at the point indicating a velocity of 3,000 ft. per min., a line is projected up to the curve *C* indicating the size of belt used and then horizontally across to the scale of horsepower, which in this instance gives 8.6 per belt. For future reference, a diagonal is drawn from this point to the zero ordinate at 5,000 ft. per min. At the point in the right-hand scale representing difference in diameters of pulley, 13 in. in this case, a line is drawn through the

diagonal scale of center distance, 21 in., and extended to the scale of loss of arc of contact, 35 deg., and then horizontally across to the curve *LMP*. In the upper horizontal scale is given the correction factor to be applied, but this is given merely for checking purposes. A line dropped down vertically from point *X* to the previously constructed diagonal and then horizontally to the horsepower column gives the net horsepower per belt, 7.8 hp., directly.

As 30 hp. is to be transmitted and one belt has a capacity of 7.8 hp., dividing gives 3.83 or four belts required for the drive.

As shown in the table below, the service correction factor for fan drives in unity.

In order to illustrate the effect when the diameter of the pulley is less than the recommended minimum, let it be assumed that the small pulley is 6.5 in. outside diameter, the large pulley, 19.5 in. outside diameter, and the balance of the data unchanged. The chart calculations will then be the same up to and including the loss of arc of contact. Referring to the table in the upper right-hand corner of the chart, the $6\frac{1}{2}$ in. diameter pulley in the column of belt size *C* corresponds to the index number 3 given in the first column of the table. This index number refers to the curves shown to the left of the table. Projecting from the point *X* of the previously established dashed line upward to index line number 1, then horizontally to the right to index line number 3 and then downward to the diagonal, as indicated by the arrows on the broken line, it is shown that the net horsepower per belt will be 6.5. This horsepower rating takes into account the loss of arc of contact and the losses arising from an insufficient pulley diameter, all based on 100 per cent belt life.

Dividing 30 hp. by 6.5 per belt gives 4.62 or 5 belts required. Thus an additional belt is required for the drive with the "undersize" pulley.

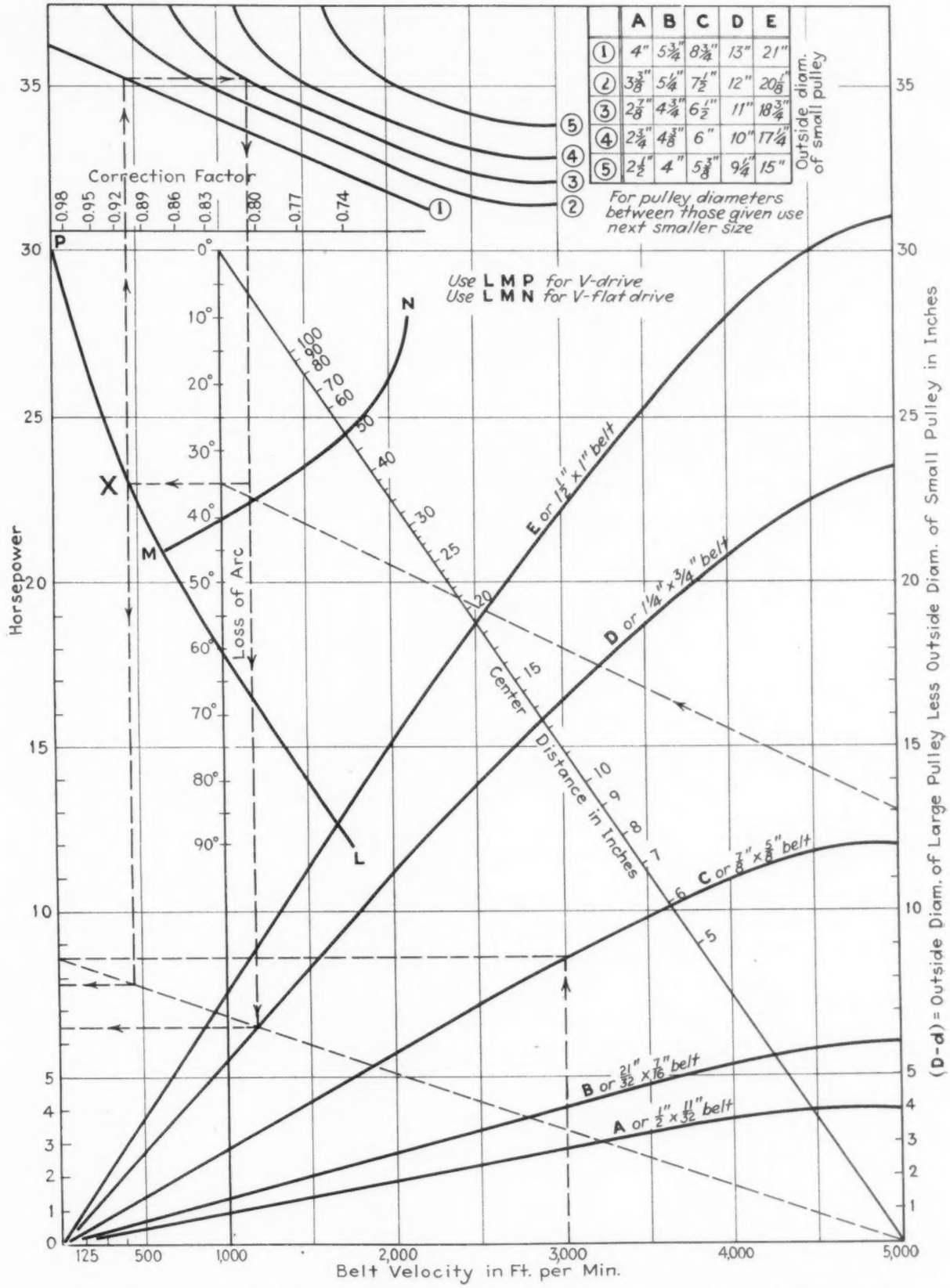
The effect of undersize pulleys on the V-belts can readily be analyzed by investigating belts that have been operating under such conditions. When passing over the small diameter pulley, the sharp bending of the belt puts excessive stresses in the cords forming the uppermost layer. After a few of these overstressed cords have broken, excessive tension is then thrown upon the remaining cords which in turn soon break. For the purpose of avoiding this as much as possible, some belt constructions use a soft rubber or rubber composition in the section of extreme compression. Sometimes rubber is used with a small amount of fabric extending through to act as a stiffening to hold the side walls in place. With a fabric cover extending over the section, the surface wear ordinarily is taken by this fabric cover.

Bearing in mind the construction of V-belts, one can readily understand why belts of standard construction will not transmit their rated horsepower when used for reverse or back-bend drives over flat pulleys. With such an arrangement the tension is thrown upon the rubber compression section of the belt which is not designed to carry tension. As a result, the belt will fail after a short time. This action is illustrated at *B* in the figure. At *C* is a section of a double V-belt wherein the cord center is placed in the neutral plane of the belt section. With this construction, only one-half of the cord section is effective when flexed over the pulley as the lower half will be in compression. When a reverse bend is made the section previously in tension will be in compression which explains why the cord section is placed in the neutral plane. The double V-belt should always be used where reverse bends are necessary.

Another successful and economical drive more recently adopted is the V-flat drive wherein a standard V-grooved small sheave is used with a flat-faced large pulley. A steel split crown-faced pulley can be used for such drives when three belts or less are involved. Only machined flat-faced pulleys should be used for larger drives. The V-flat drive is suitable only for ratios of 5:1 or greater as the loss of arc of contact over the large pulley becomes excessive when the pulley ratios become smaller.

Correction Factors for Unusual Service Conditions

Service Conditions	Type of Machine	Factor By which Normal Horsepower is to be Multiplied
Starting load light Operating load normal	Light fans and blowers Small centrifugal pumps	1.00
Light pulsations Starting or peak loads up to 125 per cent of normal	Small pumps and compressors Line shafts carrying light loads	1.10
Motors started on full voltage	Line start motors	1.20
Moderate shock or pulsations Starting or peak loads up to 150 per cent of normal	Reversing drives Large compressors and pumps Saw mill machinery (resaws, hogs, etc.) Clay working machinery (auger and plug mills, dry pans, etc.) Paper machinery (beaters, jordans, etc.) Large fans and blowers (mine fans, etc.) Large feed mills and pulverizers	1.25
Severe shock or pulsations Starting or peak loads up to 200 per cent	Mining machinery (ball, rod, and tube mills, crushers, etc.) Drives for continuous operation Small feed mills (hand feed)	1.30
Starting or peak loads up to 250 per cent. Belts run loose for shock absorption Stalling tendency	Textile machinery (spinning frames, etc.) Machines subject to frequent starting Crushers, drag lines, and other machinery driven by slip ring and series motors	1.75 to 2.00


Correction Factor
0.98
0.95
0.92
0.89
0.86
0.83
0.80
0.77
0.74
Use L M P for V-drive
Use L M N for V-flat drive
Loss of Arc
Center Distance in Inches
E or 1½" x 1" belt
D or 1¼" x ¾" belt
C or ⅞" x ⅝" belt
B or 21/32" x 7/16" belt
A or ½" x 11/32" belt
Horsepower
Belt Velocity in Ft. per Min.
(D-d) = Outside Diam. of Large Pulley Less Outside Diam. of Small Pulley in Inches
Outside diam. of small pulley
For pulley diameters between those given use next smaller size

	A	B	C	D	E
1	4"	5¾"	8¾"	13"	21"
2	3⅜"	5¼"	7½"	12"	20⅛"
3	2⅞"	4¾"	6½"	11"	18¾"
4	2¾"	4⅜"	6"	10"	17¼"
5	2½"	4"	5⅜"	9¼"	15"

A vertical line from the belt-velocity scale to the curve of belt size gives the uncorrected horsepower per belt in the left-hand vertical scale. The broken line from the right-hand vertical scale to point *X* and then down to the lower diagonal gives the horsepower corrected for loss of arc of contact. Upward from *X* to curve 1, horizontally to the numbered curve indicated by the table, and then down to the lower diagonal, gives net corrected horsepower per belt. This takes into account the loss in capacity caused by undersize pulleys

Linkages for

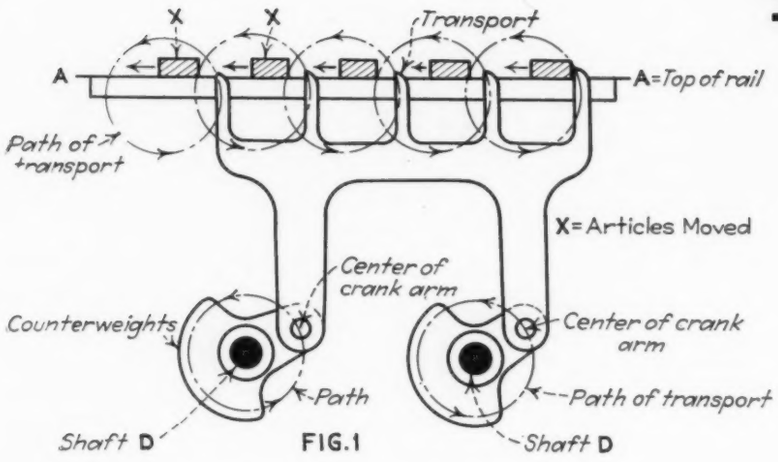


FIG.1

Fig. 1—In this design a rotary action is used. The shafts *D* rotate in unison and also support the main moving member. The shafts are carried in the frame of the machine and may be connected by either a link motion, a chain and sprocket, or by an intermediate idler gear between two equal gears keyed on the shafts. The rail *A-A* is fixed rigidly on the machine. A pressure or friction plate may be used to hold the material against the top of the rail and prevent any movement during the period of rest.

TRANSPORT mechanisms are generally used for moving material. The motion, although unindirectional, gives an intermittent advancement of the material being conveyed. The essential characteristic of such a motion is that all points in the main moving members follow similar and equal paths. This is necessary in order that the members may be subdivided into sections with projecting portions. The purpose of the projections is to push the articles during the forward motion of the material being transported. The transport returns by a different path from that which it follows in its advancement, and the material is left undisturbed until the next cycle begins. During this period of rest while the transport is returning to

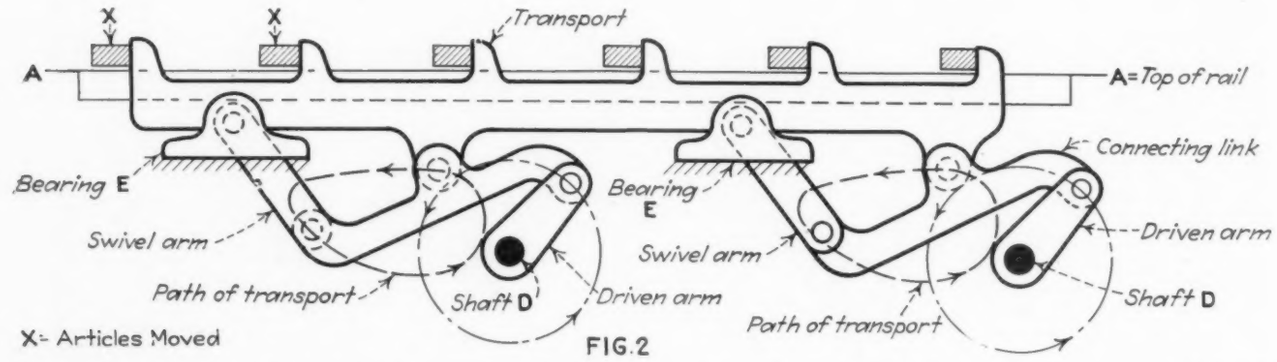


FIG.2

Fig. 2—Here is shown a simple form of link motion which imparts a somewhat "egg-shaped" motion to the transport. The forward stroke is almost a straight line. The transport is carried on the connecting links. As in design in Fig. 1, the shafts *D* are driven in unison and are supported in the frame of the machine. Bearings *E* are also supported by the frame of the machine and the rail *A-A* is fixed. The details of operation can be understood readily from the figure.

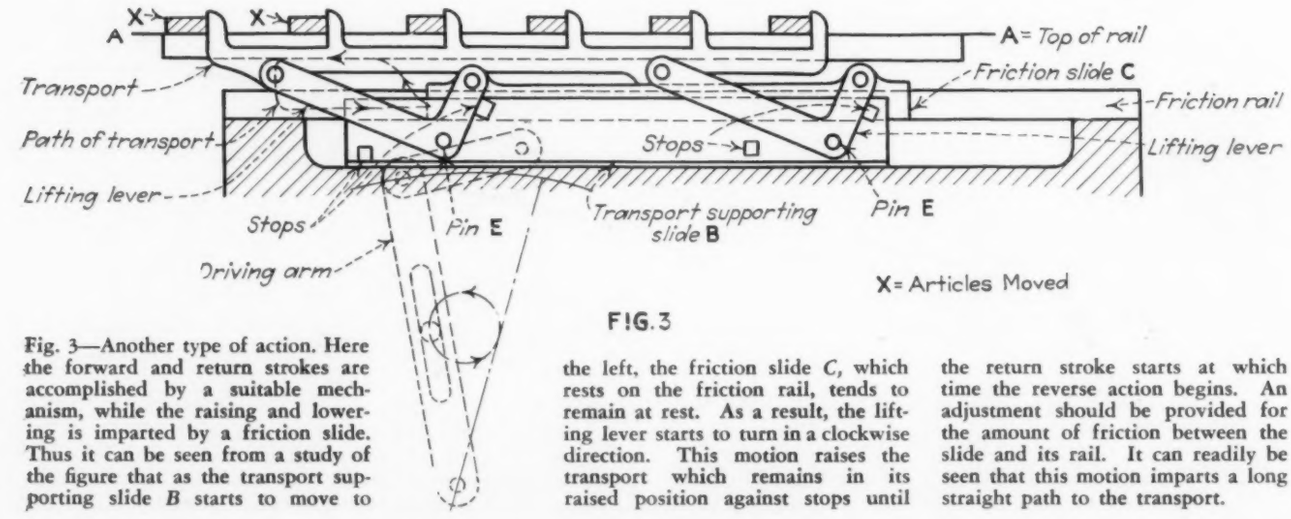


FIG.3

Fig. 3—Another type of action. Here the forward and return strokes are accomplished by a suitable mechanism, while the raising and lowering is imparted by a friction slide. Thus it can be seen from a study of the figure that as the transport supporting slide *B* starts to move to the left, the friction slide *C*, which rests on the friction rail, tends to remain at rest. As a result, the lifting lever starts to turn in a clockwise direction. This motion raises the transport which remains in its raised position against stops until the return stroke starts at which time the reverse action begins. An adjustment should be provided for the amount of friction between the slide and its rail. It can readily be seen that this motion imparts a long straight path to the transport.

Transport Mechanisms

F. R. ZIMMERMAN

The designs shown here represent a summary of typical constructions for obtaining intermittent advancement. Although they are specifically for conveying material, these same mechanisms might be adopted for other purposes

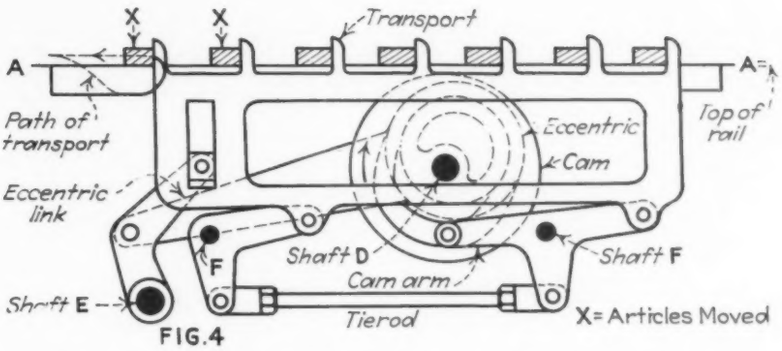


FIG.4

Fig. 4—Here is illustrated an action such that the forward motion is imparted by an eccentric while the raising and lowering of the transport is accomplished by means of a cam. The shafts, *F*, *E* and *D* are located by the frame of the machine. Special bell cranks support the transport and are interconnected by means of a tierod.

its starting position, various operations may be progressively performed.

Selection of the particular type of transport mechanism best suited to any case depends to some degree on the arrangement which may be obtained for the driving means and also the path desired. A slight amount of overtravel is always required in order that the projection on the transport can clear the material when going into position for the advancing stroke.

The designs illustrated here have been selected from numerous sources and are typical of the simplest solutions of such problems. The paths, as indicated in these illustrations, can be varied by changes in the cams, levers and associated parts. Usually the customary cut-and-try method should be used to obtain the best solution.

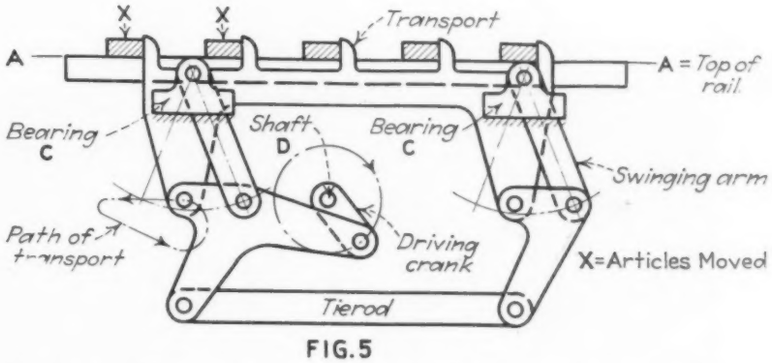


FIG.5

Fig. 5—This is another form of transport mechanism wherein a link motion is used. The bearings *C* are supported by the frame as is the driving shaft *D*.

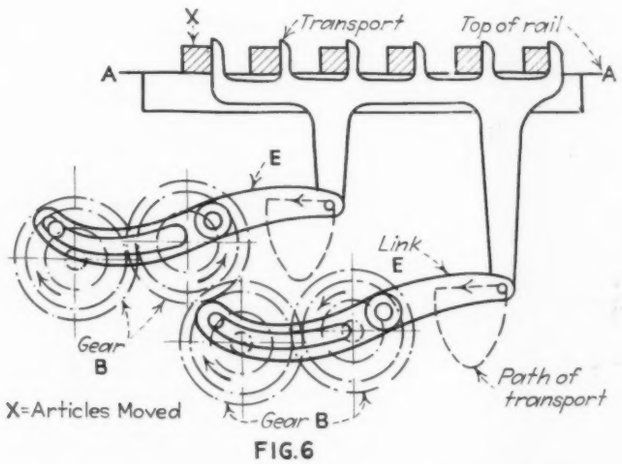


FIG.6

Fig. 6—An arrangement of interconnected gears of equal diameters which will impart a transport motion to a mechanism, the gear and link mechanism imparting both the forward motion and the raising and lowering. The gear shafts are supported in the frame of the machine.

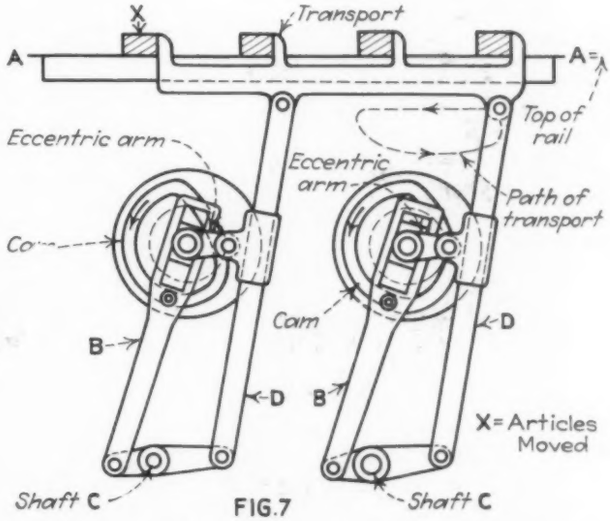


FIG.7

Fig. 7—In this transport mechanism the forward and return strokes are accomplished by the eccentric arms while the vertical motion is performed by the cams.

Case Histories in Product Design

Outlines of the life stories of three products—who originated them, who designed them, how they sold—plus the probable reasons for their success or failure

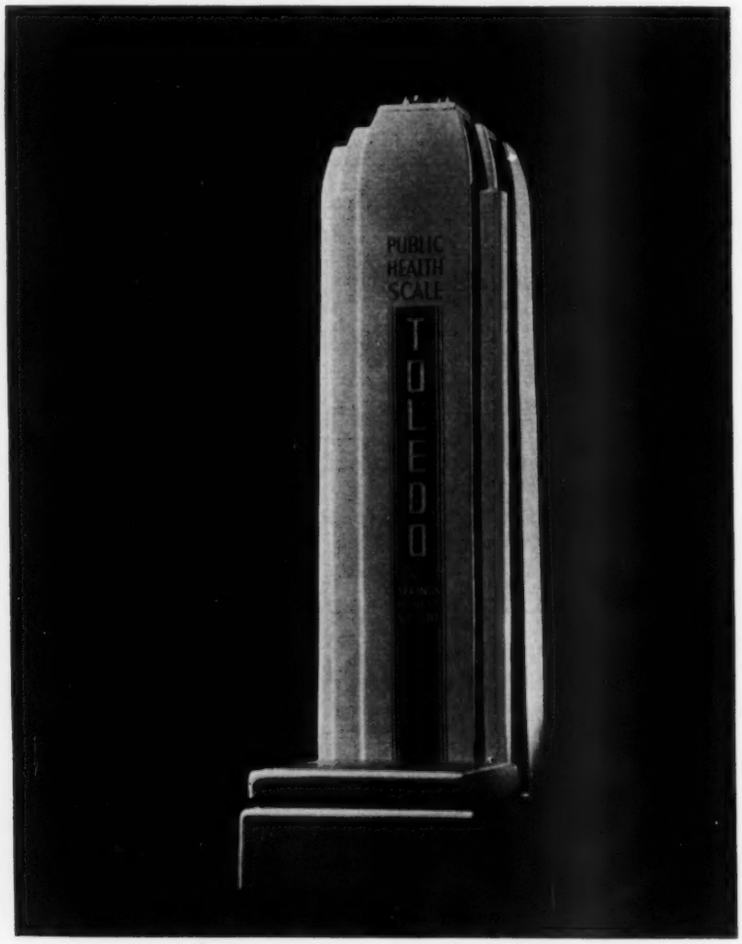
T. J. MALONEY and **GEORGE SWITZER**
New Jersey Zinc Company *Product Designer*

"When we plead for æsthetics in design we are not moved by any reformer's desire to better the world, but by an utterly selfish motive—profits." Thus, in *Product Engineering* for January, Franklin E. Brill very clearly and logically summed up the general underlying philosophy in product design, the definite places of the product engineer and the consulting industrial artist in the making of products for today's and tomorrow's market and, above all, the mental processes of the manufacturer in stimulating new ideas or revamping old ones.

Too often the consulting artist violates this philosophy. Too often the engineer lets it hold him too strictly within limits. What we propose to do in these case histories is to show the actual facts, including the conception of an idea and the resultant effects on that all-important item, profits.

Whether it was designed by the engineering staff of the manufacturer, or by a consulting industrial artist, or by both, seems of minor importance. The product is the thing. Engineers have been known to be good artists in the æsthetic senses, while artists have been known to be engineers in the engineering sense. Many products become instantaneous successes; many others—such as the automobile—are slowly changed, slowly improved, even though the research and design engineers are ready at any moment to produce a car whose conception is five years in advance of what is called the "consumer demand."

We hope this series will lead into an open forum of design case histories. To start the ball rolling we present the Toledo Public Health Scale; the Madison-Kipp Air Tools—Chipper, Filer, Grinder; the Greene Drink Mixer.



The design of the Toledo Public Health Scale received first award at the 1932 Art in Industry Exhibition

Product Case History No. 1

PRODUCT—The Toledo Public Health scale.

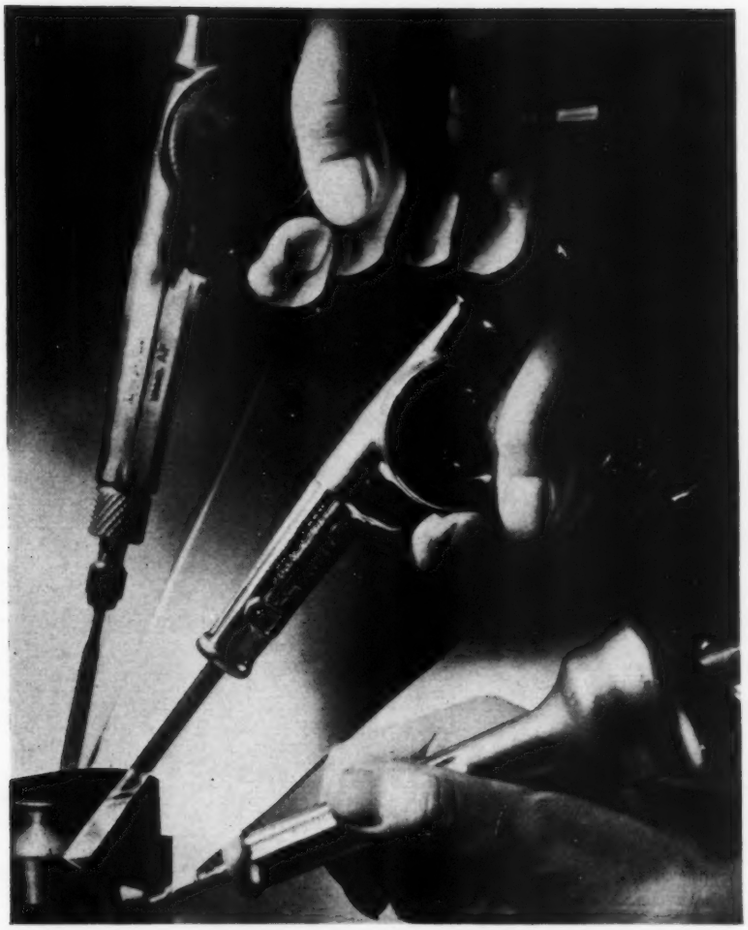
MANUFACTURER—Toledo Scale Company.

ENGINEERS—Design engineering staff, Toledo Scale Company.

ARTISTS—Van Doren & Rideout.

DESCRIPTION—The Toledo Public Health scale may be described as a small-type person-weigher made in both the coin weighing and the free weighing types. It was placed on the market on or about January 1, 1932.

The industrial artists worked from scratch with the engineers. No step was taken by one without consulta-



All three Kipp air tools—filer, chipper, grinder—are designed to mechanize slow, tedious hand operations, particularly in die finishing

tion with the other. It was decided at the outset that the new model must occupy much less space than the old. Furthermore, it was felt that women objected to the old scale because the large dial and prominent indicator revealed their weights to passersby. The new design must therefore be compact and the weight indication private. Of course, scales have been particularly well favored in color and design in the past few years, and the appearance of the scale must be both pleasing and striking.

Compactness proved to be a problem because the usual Toledo pendulum mechanism lifted the counterweights outward. To obtain a narrow column the pendulums were given a reverse swing inward across each other at the suggestion of the artists. Many other engineering problems had to be overcome to include an accurate gravity-principle machine in such small compass. Since women prefer weight secrecy, as per above, the hooded indicator has a very tangible sales appeal. The coin drops into the carrier withdrawing the shield over the indicator and making an electrical contact that illuminates the exposed chart or annular dial. Black figures on the white background give great legibility.

SALES RESULTS—Sale of the new design far outran expectations, equaling in a single year the total sales of the former design over a period of ten years. In addition, the design received first award in the 1932 Art in Industry Exhibition held in New York City. Naturally the manufacturers felt fully justified in their expenditures for design and engineering development.

PROBABLE REASONS FOR SUCCESS—Attractive exterior. Reliable interior mechanism. New scale occupies less space than the old, making it a sounder business proposition for chain stores and restaurants. Lower shipping weight. Users favor secret weighing feature.

Product Case History No. 2

PRODUCT — Kipp Air Tools—Chipper, Filer, Grinder.

MANUFACTURER — The Madison-Kipp Corporation.

ENGINEERS — Design Division, Madison-Kipp Corporation.

DESCRIPTION—All three Kipp tools were designed to mechanize slow, tedious hand chipping, filing and grinding, particularly in die finishing. They had to be of high accuracy and still stay within a very reasonable price range. On the chisel, 6,000 blows per minute are delivered. The filer has a controlled stroke speed of from 500 to 5,000 strokes a minute. The grinder operates at 40,000 to 100,000 revolutions per minute. "Brings a mile-a-minute surface speed to a half-inch wheel." All are operated by compressed air. As is normally the case, a good mechanical design proved to be attractive in appearance.

All chippers, files and grinding wheels are, of course, quickly interchangeable in their respective holders. The high precision, air turbine rotary valve mechanism of both filer and chipper provides an even flow of power when the chisel or file is pressed on the work. The pistol grip gives a natural and easy working position. The air-cushion piston eliminates vibration from the reciprocating action, providing smooth cutting strokes and a total lack of the violent vibration and strain usually associated with pneumatic tools. Both tool bodies and a large proportion of the working parts are zinc die-castings. The high rotative speed of the air grinder is accomplished by the effective use of an efficient air turbine, die-cast to the spindle and accurately balanced, mounted on anti-friction bearings. Exhaust air is utilized to blow grinding dust away from the work.

SALES RESULTS—An astonishing demand. With minimum advertising expense and a small amount of business magazine publicity, the product virtually sold itself. When continuing demand proved the value of the product, well-directed sales supervision led to additional progress in design to enter wider fields. A smaller model of the grinder selling for

approximately one-third the price of the original is just being marketed.

PROBABLE REASONS FOR SUCCESS—Despite the comment of the engineer in charge of design—"The design was an accident, the production department doesn't do what it's supposed to, and the sales department is doing a terrible job of selling"—the products entered a virgin field at reasonable prices. Since the products looked right the buying public felt that they must be right, and high-pressure sales promotion proved unnecessary.

Product Case History No. 3

PRODUCT—Greene Drink Mixer.

MANUFACTURER—Greene Manufacturing Company.

ENGINEERS — Design Division, Greene Manufacturing Company.

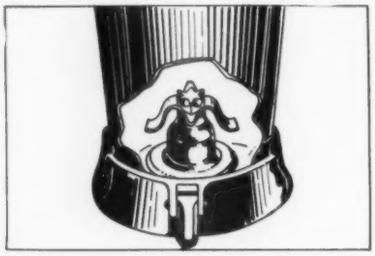
DESCRIPTION—In the Greene mixer the engineers combined a malted milk dispenser and a drink mixer in one machine. Thus it was possible to eliminate the customary bar and motor at the top, making a simpler, more compact unit. All metal parts are plated with an ample thickness of nickel and chromium. The base and dispenser rods are finished in a green or black acid-proof porcelain enamel. There are no exposed mixing parts. The simple operation of rinsing the container cleans every part that comes in contact with the drink. Mixing action takes place at the bottom of the drink container, where a bladed propeller violently agitates the liquid. The motor is started and stopped by placing container in position, and can be replaced as easily as an electric light bulb.

In all, seven aluminum or zinc die-castings were used. Quoting the manufacturer, "I might say it would be almost impossible to manufacture our product if it were not for die-castings. The element of polishing alone is nearly equivalent to the cost of raw material. The adoption of die-cast materials is very favorable to us from the standpoint of weight plus the desirable finish that is obtainable."

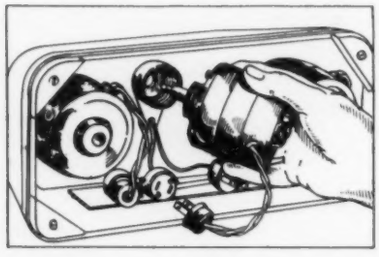
SALES RESULTS — Good, in a keenly competitive market. Two New York City chain systems are using them and a third seriously considering their installation. Manufacturer reports that markets are expanding beyond original calculations.

PROBABLE REASONS FOR SUCCESS—First of all, good design, meaning a combination of attractive appearance and general utility for the drug store soda counters. High efficiency as a food dispenser. The malted milk drops directly into the drink which is already in motion, eliminating one additional operation. Lowered manufacturing costs were obviously of great importance.

In the Greene drink mixer, the engineers combine a malted milk dispenser and a drink mixer in one machine. Motor is started and stopped by placing container in position



(Above) Each mixing cup has its own built-in agitator



(Left) The motor is as easy to mount and dismount as an electric light globe

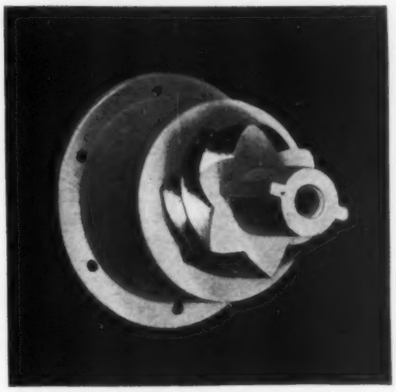
[Editor's Note—*"Case Histories in Product Design," of which the foregoing is the first installment of a series, will cover products involving both mechanical engineering and industrial art. Product Engineering readers who are interested in products involving appearance only are respectfully referred to a similar series of case histories by the same authors appearing in Advertising & Selling.*]

Getting the Most Out of Die Castings

It is hard to break away from tradition. In sand castings we have always drilled round holes and more recently broached square ones. In die castings neither is necessary, yet producers of die castings tell us that hardly a week passes that they are not called upon to figure upon designs that include drilled round holes for shafting plus setscrews and keys, or clamping lugs and what-have-you, to secure the shafting. But the die-casting art is so comparatively new that it would hardly be right to take a designer to task for forgetting that die-cast square holes for square shafts or splined holes for splined shafts are about as economical to produce as round ones.

Tradition also makes it difficult for one to remember the range of physical properties available in die castings. Development and research in alloy compositions, die design and casting equipment have been continuous, and we now have die-casting alloys available in five broad groups: tin-base, lead-base, zinc-base, aluminum-base and copper-base alloys. As just one phase of getting the most out of die castings we must keep all the alloys in mind.

The tin-base group, in which tin is alloyed with copper, antimony and lead, are used to the greatest extent in automotive bearings or in parts for soda fountains, milking machines and similar apparatus where high resistance against acids and alkalis is necessary. They also find their way into dental appliances and surgical instruments for similar reasons.



It would be difficult to manufacture this internal and external screw commercially by any method other than die casting. No machining necessary. Note holding holes and cross-key all cast in place and requiring no machining

Lead-base alloys, wherein lead is alloyed with antimony or antimony and tin, are normally used where a low-priced non-corrosive metal is required and where strength and hardness are unimportant. But both the chemical industry and the X-ray laboratory find lead-base die castings essential to their work. And lead-alloy bearings are sometimes used in preference to higher-price tin-base babbitt metal where loads and speeds are nominal.

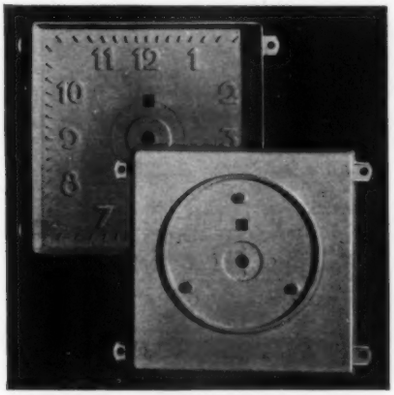
Zinc-base die castings abound in industry, the standard alloy having the following specifications in parts per hundred:

Copper	2.50-3.50
Aluminum	3.50-4.50
Magnesium	0.02-0.10
Iron	0.100 max.
Lead	0.010 max.
Cadmium	0.005 max.
Tin	0.005 max.
Impurities	0.020 max.
Zinc (99.99 per cent grade)	Remainder

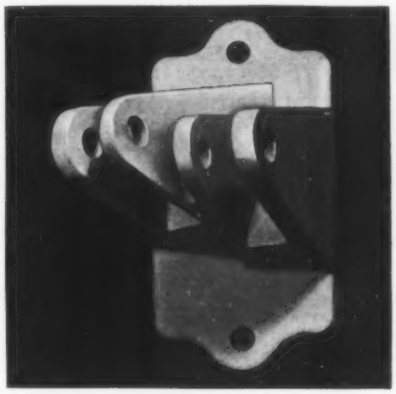
This alloy will show the following physical properties:

Tensile strength	45,000-48,000 lb. per sq.in.
Elongation	4-6 per cent
Modulus of rupture	116,000 lb. per sq.in.
Compression strength	90,000 lb. per sq.in.
Shearing strength	45,000 lb. per sq.in.
Impact strength (Charpy)	12 ft.lb.
Thermal expansion per deg. F.	0.0000155 in. per in.
Specific gravity	6.75
Hardness (Brinell)	85-90
Melting point	740 deg. F.
Casting temperature	800 deg. F.

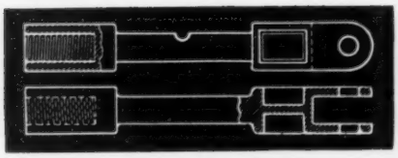
Die castings of this alloy are standard in the automotive industry for carburetors, speedometer housings, radiator trim, interior and exterior parts and hardware too numerous to mention; zinc-base parts reduce the costs and improve the quality of cash registers, washing machines, vending machines, electric motors and fans. Practically any type of commercial finish can be applied to them; they can be electroplated with nickel, chromium, copper, or the precious metals, or coated with lacquer, varnish, paint



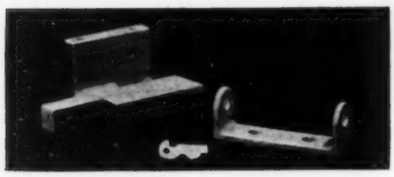
Front and rear views of a die-cast clock face with numerals, positioning ring and assembly holes all cast in place



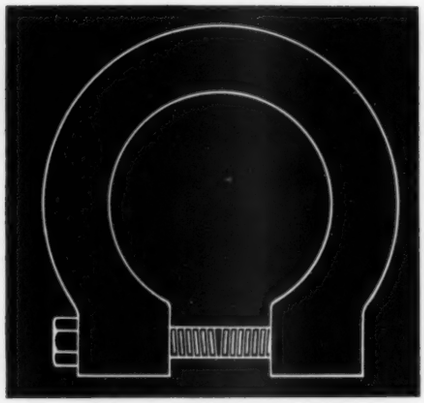
Knowing that this vari-angled part was to be die-cast, the designer did not have to concern himself with machining problems



In a die casting it is possible to include threaded sections with flat bottoms and clamping grooves for taper pins without additional machining. Note coring at clevis end for lightness and metal-saving



All three of the parts here shown would normally be designed as stampings, but the die caster found room for them in the die for a major piece and added them. Result, three additional parts for the bare cost of the metal plus a slight additional die charge



Clamping ring with steel right-and-left-hand screw die cast in place. It would be difficult to conceive how this part could be achieved by any method other than die casting

or enamel. The smooth surfaces possible on die castings require little preparation for finishing.

Of aluminum-base alloys, there are almost an unlimited number, but those used in die castings are confined practically to two groups: aluminum-copper in which the copper may be varied from 4 to 14 per cent; and aluminum-silicon, with silicon ranging from 5 to 13 per cent. A typical aluminum-copper alloy is compounded as follows: copper 7 to 9 per cent; silicon 1 to 3 per cent; iron 2.5 max.; impurities 1.5 max.; aluminum, remainder. Such an alloy will have the following physical properties:

Tensile strength
29,000-31,000 lb. per sq.in.
Elongation .....1-2 per cent
Yield point..12,000-15,000 lb. per sq.in.
Hardness (Brinell).....75-80

In percentages, a typical aluminum-silicon alloy has the following mixture: silicon 11.50-12.50; copper 0.60 max.; iron 2 max.; impurities 1.5 max.; aluminum, remainder. This alloy has the following properties:

Tensile strength
30,000-33,000 lb. per sq.in.
Elongation .....1½-3 per cent
Yield point
10,000-12,000 lb. per sq.in.
Hardness (Brinell) 70-75

Both groups of aluminum-base die castings, because of their lightness, strength, corrosion resistance and ability to take and hold a high polish, are used extensively in household utilities. Cooking utensils, vacuum cleaners and waffle griddles are frequently made almost wholly of these materials. Cameras, typewriters, motion picture projectors, metal furniture, and motor housings are all familiar examples of aluminum die castings.

In copper-base alloys we go to still higher strengths and hardnesses, a typical composition being: copper 57-59 per cent; zinc 40-42 per cent; tin 0.5-1.5 per cent; aluminum 0.1 max.; lead 0.75 max.; manganese 0.25 max. This alloy will show the following physical characteristics in pressure-die-cast condition:

Tensile strength
65,000-75,000 lb. per sq.in
Yield point
35,000-40,000 lb. per sq.in.
Elongation .....15-20 per cent
Brinell hardness.....130-140
Weight.....0.303 lb. per cu.in.

And in a copper-silicon-zinc alloy, developed by the Doehler Die Casting Co., and known as "Brastil," we find even more surprising physical properties in the die-cast condition:

Tensile strength
90,000-95,000 lb. per sq.in.
Yield point
68,000-70,000 lb. per sq.in.
Elongation .....10-17 per cent
Brinell hardness 160-180

Brastil is said also to be highly resistant to corrosion, to have good bearing qualities and to be easily machined.

Whatever the alloy group selected for a particular design, the fundamental factors of quality in die castings are strength, accuracy and uniformity. If the designer will bear in mind the present casting limits of typical die-casting alloys, as set forth in the accompanying table, he will protect both himself and his company. It is only through genuine cooperation between the designer and the die-casting manufacturer that the utmost in design and economy may be attained.

Finally, to get the most out of die castings, the designer will take advantage of the many short-cuts and economies inherent in the art, such as the examples used to illustrate this article. Die castings should always be designed to save every possible machining operation and to take full advantage of their artistic or appearance values.

[Editor's Note—*The material for this article, the illustrations and their captions were supplied through the courtesy of Louis H. Morin, chief engineer, Doehler Die Casting Co.*]

Casting Limits of Typical Die-Casting Alloys

	Tin	Lead	Zinc	Alum.	Copper
Maximum weight of casting, lb....	10	15	24	10	3
Minimum wall thickness, large casting, in.....	1/16	1/16	1/16	0.085	0.125
Minimum wall thickness, small casting, in.....	1/32*	1/32	0.035	0.050	0.050
Variation from drawing dimensions per in. of diameter or length in...	0.001*	0.001*	0.001*	0.002*	0.003*
Cast threads, min. No. per in. external.....	32	32	24	20	10
Cast threads, min. No. per in. internal.....	32†	32†	24†	None	None
Cast holes, min. diameter, in.....	0.031*	0.031	0.031	3/32	3/16
Draft per in. of length or diameter of cores, in.....	None	None	0.003	0.015	0.020
Draft per in. of length or diameter of side walls, in.....	0.0005	0.0005	0.005	0.010	0.020

* Depends on conditions. † Where cheaper than tapping.

Radially Tapered Steel Disk Springs

W. W. BOYD

The cumbersome stress and deflection equations for this type of spring tend to discourage the designer from using these high capacity compact units. The accompanying curves reduce the calculations to simple arithmetic

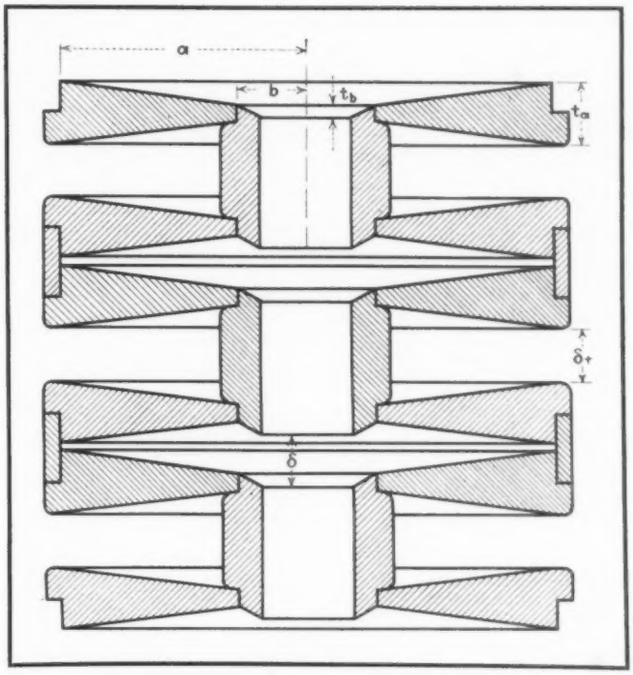


Fig. 1—In the assembled radially tapered disk spring the maximum deflection δ can be fixed by the design of the spacing sleeves

Radially tapered disk springs are made up of disks having a central hole, the thickness of the disk increasing radially outward, giving a section of a truncated wedge. In the accompanying Fig. 1 is shown an assembled spring with spacers between the disks. The maximum deflection of such springs can be controlled by varying the length of sleeve between the disks. This governs the maximum stress to which the spring material will be subjected. The beveled ends of the sleeves align them when the spring goes solid.

Radially tapered disk springs offer the advantage of materially increased flexibility that can be secured when space in the direction of the load application is limited.

In a paper entitled "The Radially Tapered Disk Spring" by Messrs. W. A. Brecht and A. M. Wahl of the Westinghouse Electric & Mfg. Co., of East Pittsburgh, Pa., published by the A.S.M.E. in 1930, known as paper APM-52-4, the following formulas are given:

$$\delta = M\frac{P a^2}{E t_b^3}$$

$$S = C\frac{P}{t_b^2}$$

The symbols used are as follows:

- δ = axial deflection of the disk, in.
- P = total axial load, lb.
- a = outer radius of disk, in.
- b = inner radius of disk, in.
- t_a = outer thickness of disk at radius a, in.
- t_b = inner thickness of disk at radius b, in.
- S = maximum stress, lb. per sq.in.
- E = modulus of elasticity, 30,000,000 lb. per sq.in.
- $\alpha = \frac{a}{b}$ = ratio of outer to inner radius
- M = deflection constant, depending on ratio α
- C = stress constant depending on ratio α.

When the ratio of the radii a/b is limited to a minimum of $1\frac{1}{2}$ and a maximum of 5, M and C are:

$$M = \frac{1}{28}\left(5-\frac{a}{b}\right)^2 + 0.04$$

$$C = \frac{1}{44}\left(5-\frac{a}{b}\right)^2 + 0.3$$

These values approximate the curves given in the above-mentioned paper, and are sufficiently accurate for practical purposes.

Substituting the values of M and C in their respective formulas, the following equations are obtained:

$$\delta = \left[\frac{1}{28}\left(5-\frac{a}{b}\right)^2 + 0.04\right]\frac{P a^2}{E t_b^3}$$

$$S = \left[\frac{1}{44}\left(5-\frac{a}{b}\right)^2 + 0.3\right]\frac{P}{t_b^2}$$

From similar triangles:

$$\frac{a}{b} = \frac{t_a}{t_b},$$

Since $a/b = \alpha$, we may write:

$$\delta = \left[\frac{1}{28}\left(5-\alpha\right)^2 + 0.04\right]\frac{P a^2\alpha^2}{E t_a^3}$$

$$P = \frac{\delta E t_a^3}{a^3 a^2\left[\frac{1}{28}\left(5-\alpha\right)^2 + 0.04\right]}$$

$$S = \left[\frac{1}{44}\left(5-\alpha\right)^2+0.3\right]\frac{P\alpha^2}{t_a^2}$$

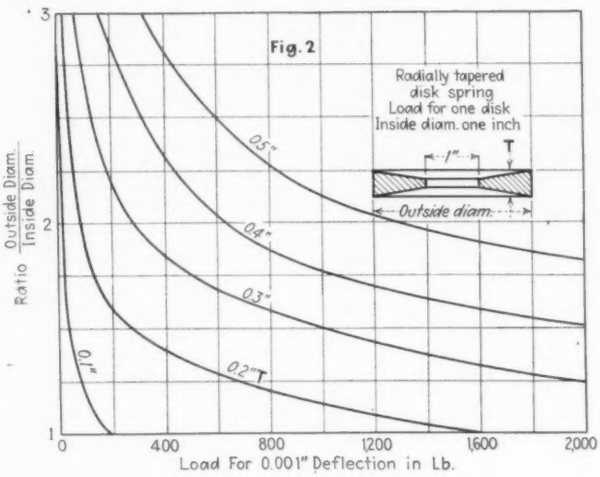
$$P = \frac{S t_a^2}{\alpha^2\left[\frac{1}{44}\left(5-\alpha\right)^2+0.3\right]}$$

If the "cut and try" method is used, much laborious work is encountered. Several trials usually have to be made before satisfactory results are obtained. But by constructing the curves the results may be obtained with minimum arithmetical work.

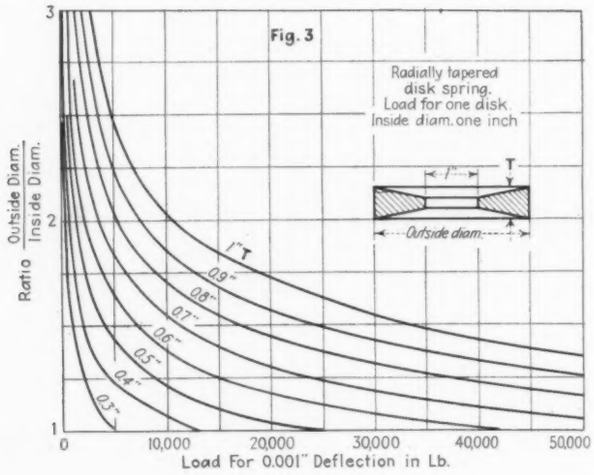
In the above equation there are five variables, but δ, S, and P are straight line functions. By assuming constant values for α, δ, and t_a, it is possible to derive curves which will give values of P for various ratio of α and outside thicknesses t_a, when the inside diameter is assumed to be unity, namely, 1 in. For inside diameters other than 1 in., the values of the load are $(1/n^2)$ times those obtained from the curve, n being the specified inside diameter of the disk being calculated.

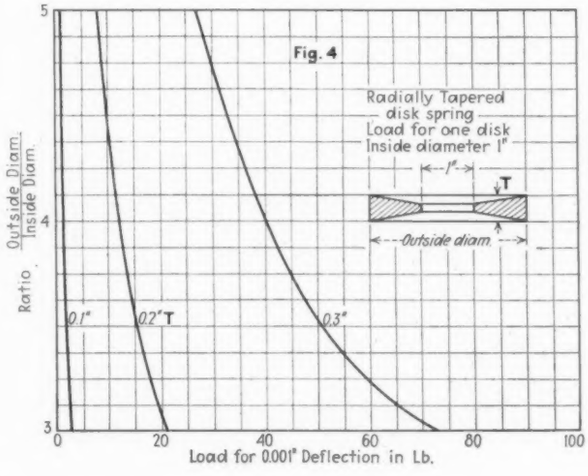
Assuming a deflection of 0.001 in. and a stress of 10,000 lb. per sq.in., curves are plotted for outside thicknesses of disk against diameter ratios and load. The accompanying Figs. 2 to 6 give the complete series of such curves for disks of various ratios of outside to inside diameter and ranging in thickness from 0.1 in. to 1.0 in. The thickness is measured as indicated in the detail drawing of the disk.

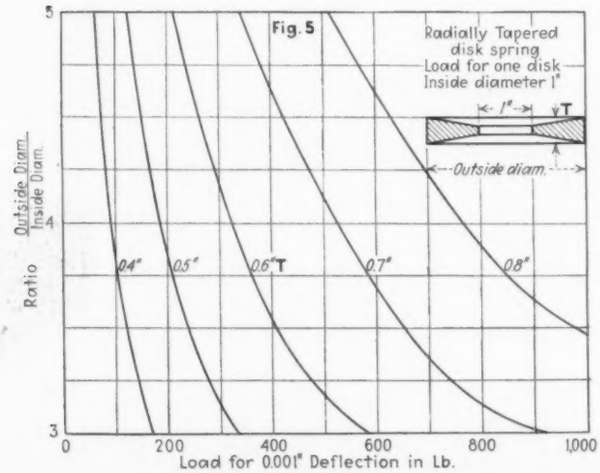
In addition to these load-deflection curves, it is possible to construct similar curves to show for different thicknesses of disks the load at which the maximum unit stress in the ma-

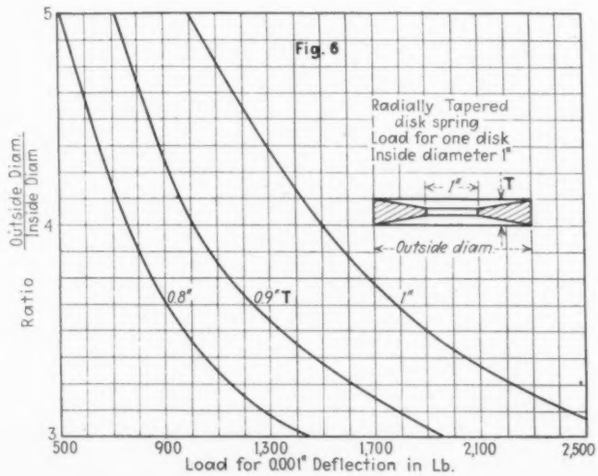


Figs. 2 to 6 — These curves give the load required for a deflection of 0.001 in. per disk. At that load the total deflection of the assembled spring in thousandths of an inch will be equal to the number of disks in the spring









terial will be 10,000 lb. per sq.in. As the maximum stress in the disk will be proportional to the load, by simple arithmetical proportion the load at any other stress, or stress at any other load, can be calculated.

A third set of curves can then be constructed to show the deflection when the inner edge of the truncated wedge is deflected as shown in Fig. 9.

Load for 0.001 in. deflection:

$$P = \frac{30{,}000\, t_a^3}{\alpha^3 a^2 \left[\frac{1}{28}\left(5-\alpha\right)^2 + 0.04\right]} = \frac{840{,}000\, t_a^3}{\alpha^3 a^2 \left[\left(5-\alpha\right)^2 + 1.12\right]}$$

For 10,000 lb. per sq.in. stress:

$$P = \frac{10{,}000\, t_a^2}{\alpha^2 \left[\frac{1}{44}\left(5-\alpha\right)^2 + 0.3\right]}$$

$$P = \frac{440{,}000\, t_a^2}{\alpha^2 \left[\left(5-\alpha\right)^2 + 13.2\right]}$$

Assuming that the tapered disk, when fully loaded, deflects until one side of the truncated wedge coincides with the corresponding edge of the disk, curves are plotted for outside thickness of disk against diameter ratios and deflection.

The data for these deflection curves is obtained as follows:

From similar triangles as shown in Fig. 1:

$$\frac{t_a}{t_b} = \frac{a}{b} = \alpha$$

Then

$$t_b = \frac{t_a}{\alpha}$$

The deflection is:

$$\delta = \frac{t_a - t_b}{2}$$

Substituting for t_b:

$$\delta = \frac{t_a - t_a/\alpha}{2}$$

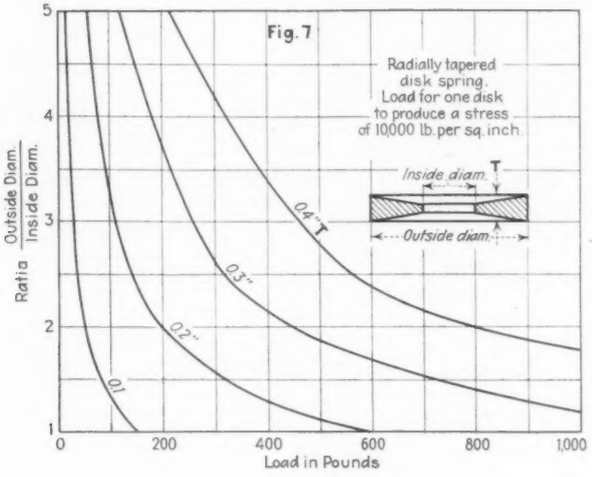
It will be noted that this deflection is constant for a given diameter ratio and independent of the diameters corresponding to that ratio.

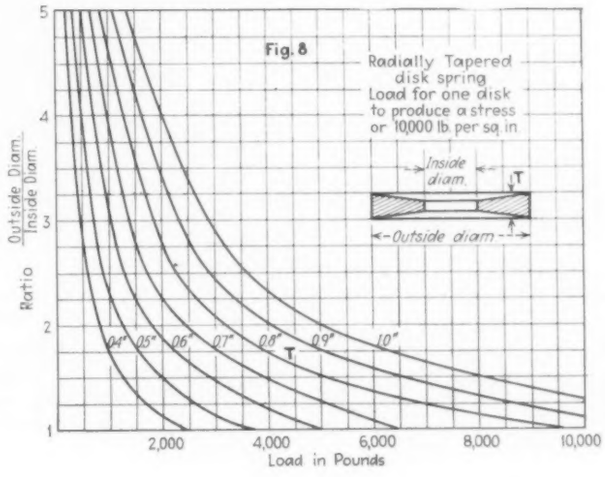
Excessive stresses may be avoided by controlling the movement of the spring through the design of the inner spring seat sleeve.

As an example of the use of these curves, assume a disk spring 0.4 in. outside thickness and a diameter ratio of 3. As shown by the curves in Fig. 5, such a disk will support a load of 170 lb. at a deflection of 0.001 in. From the curves in Fig. 7, it can be seen that the load at a stress of 10,000 lb. per sq.in. will be 465 lb. The deflection at this load will be 465 lb. divided by 170 lb. per 0.001 in. deflection, or 0.0027 in. deflection at 465 lb. load. From the deflection curve in Fig. 9, the "deflection at full load" will be 0.135 in. As a load of 170 lb. gives 0.001 in. deflection, a deflection of 0.135 in. will require a load of 170 × 135 = 23,000 lb. approximately. As the stress will be proportional to the load, the stress at 23,000 lb. load will be 495,000 lb. per sq.in.

This unit stress of 495,000 lb. per sq.in. is only an "apparent" stress. Beyond the yield point of the material the stress is no longer proportional to the elongation.

If the inside diameter of the spring happens to be 3 in., then the load at a stress of 10,000 lb. per sq.in. will be 465 divided by 9, or 52 lb., the load for a given stress being inversely as the square of the inner diameter.





Figs. 7 and 8—These curves give the load to produce a stress of 10,000 lb. per sq.in. Stress at other loads can be calculated by direct proportion

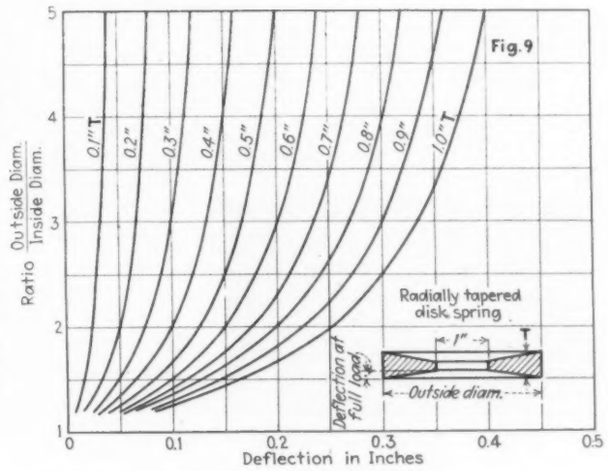


Fig. 9—As explained in the text, "deflection at full load" is not always attainable within the elastic limit of the material

COMMUNICATION and COMMENT

Engineering Small Products

LEWIS N. STEWART

To show the advantage of applying mathematics to some of the small and simple things or products, even when exact numerical values cannot be obtained, one need only consider the problem which deals with the design of an ordinary star drill such as used in drilling concrete and brick or stone masonry. It has long been the practice to make these drills from octagon bar stock slightly smaller than the nominal size of the drill. For the purpose of illustrating the application of mathematical analysis, the design of a 1½ in. drill made from 1¼ in. octagon stock and weighing 5.375 lb. will be investigated. It will be assumed that a 2-lb. hammer drives this drill and has a velocity of 25 ft. per sec. upon striking.

Let:

W = 5.375 lb., the weight of the drill
w = 2 lb., the weight of the hammer
v = 25 ft. per sec., striking velocity of the hammer
V = velocity of hammer and drill after the blow is struck
E = total kinetic energy in hammer at moment of striking

$$E = \frac{wv^2}{2g} = \frac{2 \times 25^2}{2 \times 32.16} = \frac{1250}{64.32} = 19.434 \text{ ft. lb.}$$

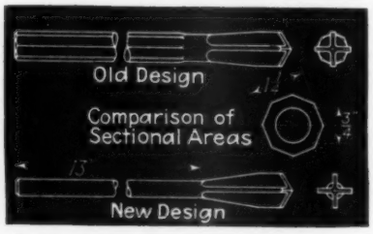
Upon striking the drill, some of the kinetic energy of the hammer is absorbed in overcoming inertia, and is consumed in internal molecular work. This last is shown by the battering of the end of the drill shank. It is evident that the amount of energy dissipated in such a manner will be some function of the ratio of the weight of the hammer and that of the drill. Therefore, it is logical to assume that the available energy for useful work will be proportional to the fraction $w/(W + w)$ of the total energy.

On this basis, using the data of this problem, the amount of energy available for doing useful work will be proportional to:

$$\frac{w}{W+w} = \frac{2}{2 + 5.375} = 0.271$$

For the sake of simplicity, it will be assumed that the available useful energy equals 0.271 × 19.434 or 5.267 ft.lb.

Thus it will be seen that 14.167 ft. lb. of energy is absorbed in overcoming inertia and is lost through internal molecular work.



By a simple change in the design of this drill its weight is reduced and its efficiency thereby increased correspondingly

Changing the design of the drill, as indicated in the sketch, the section of the drill shank can be reduced for a distance of 13 in. to a diameter of ¾ in. This will reduce the weight of the drill to 2.254 lb. instead of 5.375 lb., its original weight.

As a result of this weight reduction, the value of the fraction $w/(W + w)$ is increased from 0.271 to 0.47, thereby increasing the amount of available energy to 9.144 ft.lb., as compared with 5.267 ft.lb. in the former design. This means that the efficiency of the drill under the conditions given here has been almost doubled.

This same line of reasoning can be applied not only to drills and chisels but to any design problem where impact is involved.

Art and Engineering

To the Editor:

In your December, 1932 number, page 473, you indicate by an editorial entitled "Engineer and Artist" a state of affairs which seems to me to be fundamentally unsound, yet your editorial infers that this state of affairs is quite common. If such is the case, I would suggest that you investigate the relations between engineering art director and the various engineering departments at the Westinghouse Electric & Manufacturing Company. There I am sure you will find a very close cooperation and coordination of effort in the attempt to design a really attractive product, rather than the use of the art department as an adjunct to or a high pressure effect of the advertising and sales departments.

As intimated above, the engineering art director and his staff are part of the engineering department. As such, they are working primarily with the engineering department. As the engineers proceed with their design or development, the problem is brought to the engineering art director, with a full exposition of the purpose of the device, how it is used, by what type of people it is not only used and handled, but also observed and viewed. Rather than have the engineer's requirements limit the efforts of the artist, the latter's knowledge of the device enables him to contribute to the engineer such features of line, shape and color as will enhance the appearance of the apparatus being designed.

And, contrary to many general be-

liefs, it is not uncommon for the art director to say when an object is presented for his criticism,—"Well, that looks pretty good to me,—I don't know of anything that I would add to it." —A. J. A. PETERSON

Section Engineer
Westinghouse Electric & Mfg. Company

Broad versus Narrow Lettering

To the Editor:

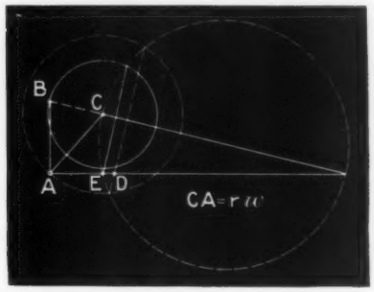
Referring to E. H. Bruce's article on lettering in the December, 1932, number of *Product Engineering,* in the writer's opinion a short, broad letter is extremely undesirable on drawings because of the great amount of room it takes up. Also, in a given space the same amount of lettering is less legible in a broad style as compared with tall, narrow lettering. This fact is recognized in the printing industries where a tall, narrow type is used for clarity, particularly where boldness is essential. Admittedly, it is slightly more difficult to make tall letters perfectly but such lettering may be relatively poor in form and yet be legible.

Mr. Bruce's method of practice is applicable by changing the guide frames from square to tall rectangles. Having become adept, the printing can be practiced further without the use of guide lines. For horizontal printing, the T-square or equivalent is used and for vertical printing the triangle or other loose straight edge serves. The straight edge is placed just under the line of printing and is used as a stop for the pencil as it reaches the bottom of all vertical strokes. The lower horizontal strokes, such as the one on the letter *L,* are made against the straight edge. This leaves only a few horizontal strokes, such as on *F* to be made without a positive guide, though the proximity of the straight edge, which remains in position until all the line of printing is completed, is of considerable aid. —W. S. BROWN.

Klein's Construction

To the Editor:

In the presentation of Klein's construction as given in the July number of *Product Engineering,* I find that there is an error in Fig. 4 on page 275. The accompanying figure duplicates Fig. 4 in dashed lines with the corrected acceleration layout superimposed in full lines. The full line construction is identical with Klein's construction, but the vector *AE,* rather than *AD,* represents the acceleration of the slider, but not measured to the same scale as the layout, as stated in the article. It can be proven mathematically that the velocity of the slider is proportional to ω, the angular velocity of the crank, while the acceleration is proportional to ω^2. In Klein's construction, the scale is chosen proportional to ω, and the length of the velocity vector gives the correct velocity to this scale. But the length of the acceleration vector to this scale must be multiplied by ω to obtain the true acceleration.



The correct layout of Klein's construction is shown by the full lines, the vector *AE* representing the acceleration of the slider

Inasmuch as both these vectors must be treated differently, a safer and more logical construction would be obtained by using a scale proportional to the physical dimensions of the mechanism, the length of the velocity and acceleration vector then merely representing coefficients, the former to be multiplied by ω and the latter by ω^2 to obtain the correct numerical values. —R. A. FREEMAN

Research & Development Department
R. Hoe & Company, Inc.

Editor's Note—A similar discussion was presented by William S. Huck, who also gave the mathematical proof. We regret that space does not permit its presentation. Mr Huck also suggested that the scale of the mechanism be made proportional to its physical dimensions and that the velocity and acceleration factors be treated as suggested by Mr. Freeman.

Recording Drawing Changes

H. L. LACKMAN

One comprehensive method for recording drawing changes uses a "Drawing Change Notice" such as shown in the accompanying illustration. After the revisions have been made on the drawing, a statement of the changes made is put on the "Change Notice" which is then blueprinted as part of the revised drawing.

In this manner, the manufacturing office and shop are given notice of specific parts to be redesigned, re-

NAME BULKHEAD, MAIN, STA. #4A. DRAWING 950-A

WHERE USED		DESCRIPTION	DATE
MODEL	ASSEMBLY		
LMI-9	-56	④ REWORK -56 AS FOLLOWS: INCREASE FLANGE RADIUS FROM 1 3/8" TO 1 1/2". ADD 2 RIVETS AT 1/2" SPACING FOR CLIP ON AFT END. FILE DOWN ANGLE ON BLK'D. TO 7/16" INSTEAD OF 9/16" CUT BACK FWD. FLGE. 9/32" INSTEAD OF 3/16". P. BROWN.	10-31-31

AUTHORIZED BY PROJECT ENGINEER'S ORDER #5.

RECOMMENDED DISPOSITION STOCK		
CLASS "3" CHANGE.	REQUESTED	R. LOCK
	AUTHORIZED	[signature]
	APPROVED	[signature]
	APPROVED	W. WALLAT

DRAWING AND PART LIST CHANGE NOTICE

Sample of a change-notice form, filled out completely

moved or added as specified in the proposed change. Furthermore, each part on the drawing that is affected bears an identifying number corresponding to the number of the change as given on the change notice. A circle is drawn around this number to indicate that it is a change number. This method affords an easy identification of the work to be done.

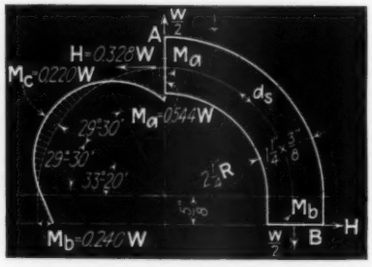
The class of the change refers to the type or character of the change to be made. This numbering follows a definite system. For instance, a Class 2 change signifies that the revision is to be effective at once, existing parts are to be scrapped, and where possible, the standard parts are to be salvaged. A Class 3 change means that the change is to be effective at once but that the parts can be reworked as specified, a Class 4 change means that the revision is to be effective after the parts on hand, in process or on order have been used. If the change involves a completely new part, previously non-existent, it is identified as a Class 5 change, which number also signifies that the revision is to be effective at once.

The change notice is written for each separate drawing affected. When the revision of a drawing involves several classes of changes, a separate change notice is made for each class of change, each change notice bearing the same date and witness number as the alteration column on the drawing.

All parts, details and processes affected are specifically mentioned in the statement of the changes. To make the record complete on the change notice, the statement of the change reads "Changed from —— to ——."

Stresses in Curved Bars

Here are two solutions to the problem of the stresses in a curved bar as presented by F. D. Faulkner in the December *Product Engineering*. It will be noted that there is considerable difference between the results obtained. Next month will be given the result of a test on such a bar and a discussion of the stress calculations.



To the Editor:

It is quite a difficult problem to calculate the stresses and deflections in a semi-circular bar such as shown by Mr. Faulkner. In this particular problem, because of the relatively high moment of inertia and area of the 2½ in. square connecting bar, the deformation caused by the moments and forces resulting from loading the ring can be assumed to be negligible without introducing appreciable error.

Symmetry about the vertical axis makes it permissible to consider only one-half of the ring *A-B*. Equilibrium is established by the vertical forces $W/2$, horizontal force H, and the end constraining moments M_a and M_b, as indicated in the accompanying figure. The general equation of bending moment for any section can then be written in terms of W, H and one of the constraining moments, for example M_a. To solve for H and M_a requires two simultaneous equations each containing both unknowns.

Since the two ends A and B are fully constrained, the total change in slope of elastic curve over the half ring A-B is 0. Hence, i the total change in slope will be:

$$i = \int \frac{M}{EI}\, ds = 0$$

Also the horizontal deflection component over the half ring A-B must be zero, giving for the total horizontal deflection component:

$$h = \int\int \frac{M}{EI}\, ds\, ds + \int \frac{T}{EA}\, ds + \int \frac{F}{NA}\, ds = 0$$

The component of W and H producing tension on a normal plane is denoted by T and the component producing shear denoted by F. It should be noted that tension and shear produce sufficient deflection to warrant consideration.

After M_a and H are known, the vertical deflection d can be found from the following equation:

$$d = \int\int \frac{M}{EI}\, ds\, ds + \int \frac{T}{EA}\, ds + \int \frac{F}{NA}\, ds$$

Also the moment at any section can be found and is a maximum at A. A complete calculation of the solution gives the following results:

$$d = \frac{16.7\ W}{E}$$

For W = 1450 lb., d = 0.001 in.)
W = 2,500 lb., d = 0.0017 in.)

The maximum combined stress occurs at the top where the load is applied.

$$f = 5.92\ W$$

For W = 1450 lb., f = 8,600 lb. per sq.in.
W = 2080 lb., f = 12,300 lb. per sq.in.
W = 2500 lb., f = 14,800 lb. per sq.in.

Since the stress is largely the result of bending, it will be possible to increase the load to 2,500 lb., before any permanent set will take place. This is 20 per cent above the elastic limit. Stress and deflection formulas will not apply with accuracy above 2,500 lb., since these formulas are based on elastic theory. As the load approaches the breaking point, the bar will be sufficiently distorted to make deflection and stress predictions inaccurate.

The stress distribution in curved bars, as caused by shear, direct stress and bending moment is set forth in detail in an article "Stresses in Heavy Helical Springs," written by the author and published in the Journal of the Franklin Institute, June, 1930.

—Elmer Latshaw.

To the Editor:

In the problem of the stresses and deflections in the closed hook, since the radius of curvature does not exceed ten times the depth of the section, the formulas for ordinary bending do not apply. The problem is analogous to that of determining the stresses in a chain link. The maximum stress evidently occurs on the inside of the ring at the section Z and is obviously a compression stress. The following equations can be used for calculating this stress:

$$f = \frac{W}{2\,(B - A)} \left(\frac{a + 2r_1}{a + \pi r_1} \right) \left(\frac{B}{A} - \frac{r_1}{r_1 - c_1} \right)$$

$= $ maximum stress

A = area of cross-section

$$= \frac{5}{4} \times \frac{3}{8} = 0.46875$$

B = modified moment of section

$$= r_1 \int \frac{dA}{r_1 + y}$$

$$dA = bdy$$

Since b is a constant:

$$B = r_1 b \int \frac{dy}{r_1 + y}$$

$$= r_1 b \log_e \left(\frac{r_1 + y_1}{r_1 + y_2} \right)$$

$$= \frac{23}{8} \times \frac{3}{8} \log_e \left(\frac{2.875 + 0.625}{2.875 - 0.625} \right)$$

$$= 0.4763$$

Substituting in the stress equation, we get:

$$f = -16{,}990 \text{ lb. per sq.in.}$$

The negative sign indicates that the stress is compressive. The deflection in the direction of the load is given by:

$$\delta = -\int \frac{M}{EI} \times ds$$

Using the relations $ds = -rd\alpha$, where α is the angle from the vertical to the radius vector,

$$y = r_1 \cos \alpha,$$
$$x = r_1 (1 - \sin \alpha)$$

$$M = \frac{Wr_1}{2} \left(\frac{a + 2r_1}{a + \pi r_1} - \sin \alpha \right)$$

$$\delta = -\int_{\frac{\pi}{2}}^{0} \frac{Wr_1^3}{2EI} \left[\frac{a + 2r_1}{a + \pi r_1} \left(1 - \frac{\pi}{2} \right) + \frac{\pi}{4} - 1 \right]$$

For the load given it is useless to apply this formula since the stress induced exceeds the elastic limit and no formulas will hold above that point. For the deflection up to the elastic limit:

$$f = KW = 16{,}990 \text{ lb. per sq.in.}$$

$$K = \frac{f}{W} = \frac{16{,}990}{1{,}450} = 11.71$$

$$W = \frac{f}{K} = \frac{12{,}300}{11.71} = 1050 \text{ lb. (at elastic limit)}$$

$$\delta = \frac{-1050 \times 2.875^3}{2 \times 25 \times 10^6 \times 0.061} \left[0.68087 (-0.5708) + 0.7854 - 1 \right] = 0.00493 \text{ in.}$$

The maximum load which the link will support is equal to:

$$W = \frac{f}{K} = \frac{35{,}650}{11.71} = 3{,}045 \text{ lb.}$$

One important thing should be noted. The formula for f contains a fraction $\frac{W}{2(B - A)}$. This part always has a large value, the remainder of the equation being nearly unity. The difference $(B - A)$ is a function of the physical dimensions of the bar, hence a small error in measurement or calculation will result in a very large difference in stress calculation. This emphasizes the necessity for using accurate tables of natural logarithms. —H. F. Bariffi.

DEFLECTIONS

CREEP PREVENTION

Small additions of molybdenum to 0.40 to 0.50 per cent carbon steel are recommended by *Steel* to prevent creep at high temperatures. Such castings will be found useful in values for superheated steam lines, and in tongs, hooks and small parts for handling hot materials.

RARE METALS LESS RARE

Dr. Frank H. Driggs and his associates in the Metallurgical Research Laboratories of the Westinghouse Lamp Co., are reported to have developed a new method of electrolysis for obtaining such rare metals as uranium, thorium and tantalum in the pure state. Being continuous and therefore cheaper, this new process is expected to make rare metals less rare, thereby extending their industrial uses. To extract thorium for example, a cathode is inserted in a liquid bath containing equal parts of sodium and potassium chloride to which has been added a quantity of potassium thorium fluoride. After powdered thorium has collected to a thickness of about 1 in. on the cathode, it is removed and replaced immediately with another cathode, the feature which makes this process continuous.

BRIGHTER AND BETTER RED LEAD

According to the U. S. Bureau of Standards, a finer grade of red lead is available. Its texture is soft, and its bright red-orange color is lighter than the average red lead. Where 32 lb. of the latter must be used to the gallon of linseed oil to make a paint of good brushing properties, the new lead requires only 16 lb. to the gallon. Since the new paint dries to a slick, glossy finish, the Bureau warns that if it is to be used as an undercoat it must contain turpentine (or mineral spirits) and drier to cut down gloss.

PLASTIC NAIL FILES

Up at General Plastics, Inc., they were testing some Durez molding compound and someone dropped a manicuring file in the bottom of the mold. When the test-piece was ejected, there was the imprint of the file in the bottom of the piece. The resultant plastic file is naturally softer than metal, but harder than finger nails. G-P is now thinking of producing emergency files molded into jar caps and cosmetic containers, or possibly in pencil barrels, on the theory that regular files are never around when they are needed.

SCENTS IN DESIGN

We have all heard of the automobile salesman who sprayed the interior of his car with perfume to give fair customers a suggestion of luxury. Now comes Donald A. Laird, director, Psychological Laboratory, Colgate University, with a report that lilac-scented rubber bands for office use are available. Though humor can be extracted from the thought that the typical and rather disagreeable smell of rubber can be completely drowned by the fragrance of new-mown hay, it is a hard-boiled fact that the use of aromatics to increase business and break down sales resistance has almost endless possibilities.

Trends in Automobile Design—II

As Revealed By the Passenger Cars Exhibited at the National Automobile Show, Grand Central Palace, N. Y. C.

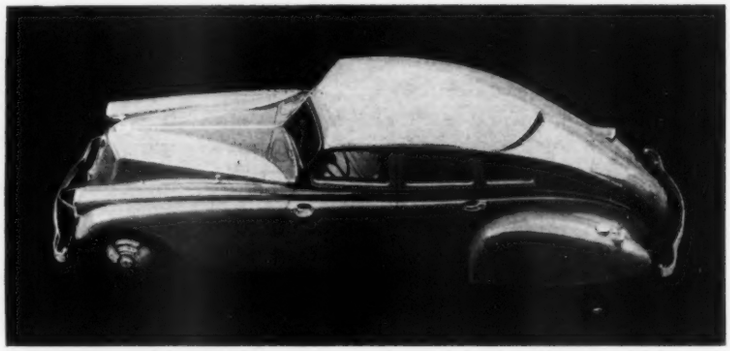
Speaking at the annual Show Dinner of the Metropolitan section of the S.A.E., an English engineer voiced a query which persists from year to year: "Is there a market in America for lighter, more efficient cars which will give from 40 to 50 miles per gallon of gasoline?" He might have voiced a parallel query: "Is the American automobile public becoming 'conscious' of operating expenses rather than first cost?"

If automotive designers had the first question in mind while preparing their higher-priced 1933 models, they answered it generally in the negative; if the second question, they answered it in the affirmative. Though as previously reported (*PE*—Jan. 1933) the trend of low-price automobile design is divided toward larger and smaller cars, the trend in the higher-price group is definitely in line with a "bigger and better" philosophy.

While an undivided trend is apparent in mechanical design, it should be noted that the various makers group themselves roughly in the two classes as regards body design. Both conservatives and radicals go in for so-called streamlining, but the conservatives cling to the fundamental outlines of previous models. Possibly the radicals are more conservative than the conservatives because their designers have seemed to feel free to copy the "silhouette" and "skirted" fenders of one of the more popular of last year's offerings. To the outside observer the progress toward the ultimate "tear-drop" car seems to be halting if not actually halted.

Snowplow Front Ends

Whatever the shape and construction of the actual radiator, designers have committed themselves definitely to the V-type or "snowplow" front end. Whether sloping or more or less upright, the V is there. Automatic shutters, on the other hand, are conspicuous by their fewer numbers, a return having been made in several cars to thermostatic control of water circulation.



Most stream-lined car at show—special Pierce-Arrow with 115 m.p.h. maximum speed, mounted on a 12-cylinder, 175 hp. chassis. Only five of this model will be built

Engines reflect refinement rather than revolution in design. Compression ratios have again been increased, the range being from 5.5 for the Studebaker and Dodge to 6.2 for the Hudson Six with aluminum composite head. Again the passenger-car designers have borrowed from the truck designer in the adoption of exhaust valve inserts. The Chrysler group of cars uses a tungsten-chromium-nickel alloy; Franklin uses Ni-Resist; Willys has adopted high-speed steel. Pierce Arrow, on the other hand, uses cast electric-furnace chromium-iron for its 12-cylinder engine cylinder blocks.

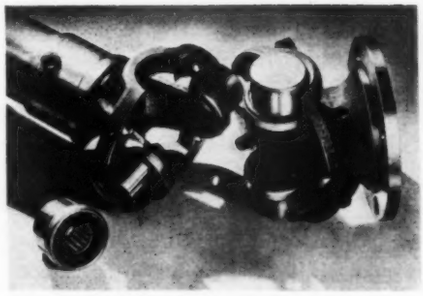
With higher engine speeds in mind, designers seem to have made a general effort to lighten reciprocating parts. Though piston practice was thought to have been standardized long ago, several makes carry four piston rings instead of three, and one make carries five. Several manufacturers are using $\frac{1}{8}$-in.-wide compression rings, thereby reducing friction. The desired unit pressure can be obtained with less outward radial pressure than in the case of a wider ring, and it is said that the narrow rings do not wear barrel-shape on the periphery as a result of rocking action in the piston.

One of the most startling departures to be found in engine design is the use of an electric furnace cast-iron alloy in the camshaft of the Essex Terraplane. It is made in a combination mold of metal and sand, the working surfaces of the cam being chilled.

In the Dodge engine, the connecting rods, shafts, pistons, rings and pins are matched in sets in which the several variations in weight are held down to 0.32 oz. But in general there is an increasing tendency to hold close weight limits in the individual parts and thus to do away with selective assembly. Though no sample was available at the Show, it is said that Packard has a new design of connecting rod in which the cap is provided with fins for additional stiffness and increased heat-radiating capacity. The seats for the bolt heads are stiffened by extending the flanges of the shank on beyond the seats. In the new Pontiac eight-cylinder engine, which by the way is now a Straight 8, the connecting rod shanks are of cruciform section and are provided with lugs near the big end

from which metal can be removed for equalizing the weights of all rods and for shifting the center of gravity of individual rods.

As nearly as could be observed, there is nothing strikingly new in engine lubrication systems. But lubricating features are making their way from the high price to the lower price cars, several of the latter having rifle-drilled connecting rods for pressure lubrication of the piston pins. Oil temperature regulation has been dropped by one maker, the explanation being that the new thin-walled steel-backed bearing bushings radiate heat more effectively and do not require temperature regulation. More cars are equipped with floating oil-intakes whereby oil is pumped from the surface of the supply in the sump rather than from the deepest point where sludge accumulates. Crankcase ventilation is continued by all makers who have previously adopted it. More manufacturers seem to be paying attention to oil leakage on garage floors, and motorists in general will say "amen." For



The Studebaker universal joint is an example of the trend toward "needle bearings." It should be noted that similar bearings are also being applied to steering gears and transmissions with good results

example, in the new Pontiac the flange on the crankcase pan is corrugated for greater stiffness and better oil sealing. A check valve is also provided in the same car in the return passage from the oil-slinger groove at the rear engine bearing to prevent oil escaping through this passage when the car is on a steep grade.

Several makers have sought to improve their ignition through the use of larger spark coils. There also seems to be a definite trend toward larger generators and batteries, or at least greater capacity ones with 15 instead of 13 plates. The 14-mm. spark plug continues to replace the 18-mm. size. The smaller plug seems to have less tendency to overheat, and thus its use is favored on high-compression motors.

Engine mounting seems still to be in a state of change. As previously reported, Willys has taken a license on Chrysler floating power; Chevrolet has developed a mounting of different design but similar characteristics; Continental has a mounting all its own as illustrated. Cadillac and some of the other high-price cars simply utilize more rubber in the support to give softer mounting. Buick has a five-point mounting, two at the front spaced as far apart as the two at the bell housing and a fifth one at the rear of the transmission.

Downdraft carburetors and air cleaners are now the rule rather than the exception. Probably more than half the models on the market have the carburetors above the engine. "Loading" of the manifold and valve pockets which might result from excessive choking of this type of carburetion is now prevented on many models by automatic thermostatically-controlled chokes. A thermostat of the well-known bi-metallic strip type and a vacuum cylinder are used to control the choke valve. In some cars, notably Pierce-Arrow, Packard, Oldsmobile, Studebaker and Reo, the automatic choke is supplemented by an automatic idle adjustment whereby the engine idling speed is greater when the engine is cold than when hot. Manual heat control of the intake hot spot is also being supplemented by thermostats on certain cars.

In this year's cars both starting and stopping seem to have received more than their share of the designer's attention. Chevrolet utilizes a vacuum control diaphragm to connect the starter mechanism with the accelerator rod when the engine is not operating and no suction exists in the intake manifold. To operate the starter, one depresses the accelerator. When the engine takes hold, the manifold suction disconnects the starter control by deflecting the diaphragm after which the accelerator pedal operates on a carburetor throttle in the usual way.

A different principle, however, is used on the Chrysler six-cylinder models and on the De Soto. A movement downward on the accelerator pedal as far as it will travel shifts the starting motor pinion into mesh, closes the starter switch and opens the throttle to the desired adjusted amount for starting. When the engine starts and with pressure on the accelerator pedal momentarily released, the vacuum-operated diaphragm in the selector causes the pedal to function then only as a throttle. This throttle connection remains as long as the engine continues to run. Still another principle is utilized on the Chrysler 8-cylinder cars, wherein a solenoid actuated by the accelerator pedal shifts the pinion into mesh, closes the starter switch, and the starting motor starts the engine. Buick and Pontiac use a button which when pressed energizes a solenoid mounted on the starting motor housing which in turn engages the pinion. The starter cannot be engaged unless the ignition switch is turned on.

Wider Brakes

As to stopping there is a trend to wider brakes with smaller diameters due to the trend toward smaller wheels. For instance, the Packard Company has changed from 16x2 to $15\frac{1}{8}$x$2\frac{1}{2}$ in., and from 15x$1\frac{3}{4}$ to 14x$2\frac{1}{4}$ in. The use of cast iron has permitted smaller diameter brakes in spite of higher driving speeds. Lining manufacturers have found it necessary to furnish material having a greater wear factor and less "fade" at high temperatures. A coil spring surrounds each Cadillac drum to give additional cooling surface and to absorb any noise produced by vibration in the drum.

All Cadillac models also employ booster brakes with a consequent shortening of the brake pedal on the Eight and on the La Salle. The Auburn 12, the Hudson 8, and the Chrysler Imperial Custom also utilize booster units. A dash control allows a varying degree of boosting for different road conditions. The hand brake operates on the rear wheel through the same shoes of the hydraulic system. Pierce-Arrow is using the Stewart-Warner Servo system with a single cross shaft on the front axle and one on the rear axle, with a single cable connection forward and to rear from the unit. A treadle not unlike the accelerator pedal is used instead of a conventional brake pedal owing to the small movement and the light pressure required.

As to power clutch control or automatic clutch control, it appears that

in its original form the clutch sometimes took hold too suddenly. If the accelerator is depressed suddenly from the idling position the vacuum in the inlet manifold, which holds the clutch in the disengaged position, is relieved very quickly, which has the same effect as removing the foot from the clutch pedal too rapidly. This has now been remedied by the provision of an additional member, a pendulum-type cushion valve, which controls the air inlet to the power cylinder. Under normal conditions the pendulum occupies a vertical position and the valve is then open, allowing atmospheric air to be drawn into one end of the cylinder as the piston therein is drawn toward the other end by the inlet vacuum. But if the car tends to start with a jerk, the pendulum instantly springs around its pivot and closes the air valve, thereby greatly reducing the speed with which the clutch engagement is completed.

Although automatic clutch control is optional on Cadillac, La Salle, Hupmobile and Oldsmobile, it is standard equipment on all Chrysler and Buick lines and on certain Hudson and Stutz models. Standard equipment on Packard last year, it is offered as optional this year.

For foot operation of the clutch, Packard and Oldsmobile are both using anti-friction bearings on the pedal shaft and certain other moving parts. Clutches themselves have also been improved, one clutch having been so redesigned that all adjustments can be reached from the transmission side of the flywheel.

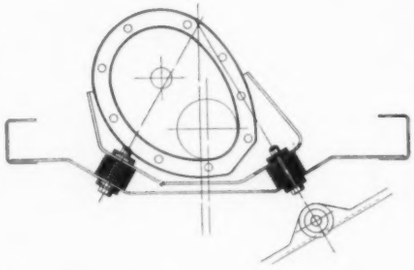
Helical Gears Standard

In transmission practice, helical gears are very much in the ascendency, as for example in the Chrysler line where all transmission gears including the reverse idler are of the helical type. The countershaft drive and second speed gears are in constant mesh, and the light clutch member reduces shifting effort and lever travel. The clutch teeth on the gear hubs are alternately under-cut, and in the sliding clutch member alternate teeth are entirely omitted. The low and the reverse gears slide on a helical spline.

It is noteworthy that the plain bearing countershaft is now obsolete. In some instances a countershaft and reverse idler are mounted on needle bearings to take the place of bushings. The 1932 Buick transmission is continued, except that the detents in the shifting mechanism have been changed from plungers to balls eliminating a tendency to stick. The gear shift lever ball on both the Buick and the Graham is made of soft rubber.

The "straight-through" muffler seems to have been generally adopted, but where one manufacturer does away entirely with sound-absorbing material in the chamber surrounding the central pipe, another uses two different kinds of such material. In the Buick, the change to the "straight-through" type is said to have resulted in a reduction in back pressure from 5 to 1½ lb. per sq.in.

Though spring and axle practice would seem to have been standardized this long time since, both Dodge and Plymouth adopt tubular front axles in the same year that Franklin switches from tubular to I-section drop-forged. Franklin has also gone to semi-elliptic springs on two of its models which makes the semi-elliptic spring standard almost unanimous. Continental, on the other hand, uses transverse front springs on at least two of its new models and a new form of double quarter-elliptic springs for rear mounting.



Detail of Continental's "winged power" front engine mountings showing projected axis of two rubber-cored springs intersecting at point on neutral axis

As regards rear axles, the adoption by Pierce-Arrow of worm drive for its eight-cylinder models is worthy of note. Nash continues to use worm drive, and Packard continues the hypoid-gear drive axle. It is also worthy of note that in one car the rear seat pan is discontinued over the rear-axle center housing and replaced by a rubber diaphragm, so that when the clearance is fully taken up, the rear axle, instead of striking the rigid seat pan, contacts with rubber.

In appearance, drive shafts and universal joints seem to be standard, but on closer examination one finds that needle bearings are being used in the universal joints of several makes. On the Plymouth, the needle bearing universal joint at the forward end is provided with trunnions sliding in a "pot-type" outer member, which eliminates the need for a sliding splined fitting.

More Needle Bearings

Needle bearings have also found their way into steering gears, being used probably not so much to reduce steering effort as to retard development of slack. Three-spoked steering wheels are almost universal, and kick shackles have been introduced on several additional cars this year.

Although "ride control" is still continued on several makes, several designers have gone over to what is known as inertia control of shock absorber action. With the ordinary shock absorber the action is dependent entirely on the relative motion between the frame and the axle which is slowed down and reduced by the shock absorber. The new inertia-controlled shock absorber differentiates between absolute motion in space of the frame and of the axle respectively. As long as the frame continues at about the same elevation there is very little shock absorber action, even if a wheel drops into a chuck hole and thereby increases the distance between frame and axle rapidly. But if the frame is thrown upward this action is at once checked by the shock absorber, the relief valve of which is closed by a spring-supported inertia weight which naturally tends to remain behind when the frame starts to move upward violently.

The Franklin 12 and Studebaker are equipped with ball-bearing shackles. The Hupp 6 shackle is a combined rubber and threaded-bolt type. The Chrysler Imperial Eight springs are rubber-bushed at each end with the addition of "oil-less" bronze bushings at the rear end of each spring.

The X-frame is as nearly a standard design as automobile designs can be. In the Essex Terraplane the bottom pan of the body has 19 points of attachment to the frame, including several points on the X-cross-member, thus tying the frame structure and the body together to form a comparatively rigid unit. On this basis an unusually light frame section can be used. Auburn, the originator of the

X-frame, has added to it an "A-frame" wherein the X-cross-member channels are continued diagonally forward immediately after being secured to the side rails. These terminate at the center of the front cross-member and provide an excellent engine mounting. In general, wheelbases and treads have been virtually unchanged.

Though the doughnut tire has yet to make its appearance as standard equipment on any car, the whole trend seems to be to small rims. The 17-in. rim dominates the field with the Dodge and the Terraplane 8 using 16-in. rims.

It would hardly be the province of this article to go into the numerous redesigns and refinements of the accessories and body hardware used on the various cars. One thing, however, should be mentioned. The drum-type of speedometer is almost as extinct as the hoop-skirt. Large-diameter dials, big figures and pronounced graduation marks tend to make the present instrument more legible. Color schemes are in the main conventional. "Pearl-luster" finishes seem to have passed with 1932.

In conclusion it should be added that just one thing was missing at this year's show—a sensational innovation in the nature of four-wheel brakes or free-wheeling. It may even be that free-wheeling is passing out of the picture, for no maker really features it, and at least one maker is now considering it as an extra. Though it is to be regretted that some designers saw fit to copy other men's designs far too closely for comfort and conscience, automotive designers in general should be congratulated on their achievements in a year of the tightest budgeting and the lowest prices in history.

[Editor's Note—*Grateful acknowledgment is made to P. M. Heldt and Austin M. Wolf for their thorough reviews in Automotive Industries and S.A.E. Journal respectively.*]

"New Things—Developments During the Depression"

Engineers and manufacturers are busy with new ideas and new products, the Patent Office is speeded up, and radically new processes, machines and materials are being offered to a waiting public, pointing the way back to re-employment and the end of the depression, according to Dr. H. C. Parmelee, vice-president and editorial director, McGraw-Hill Publishing Co., who addressed the Science Forum of the New York Electrical Society, Thursday evening, Jan. 12, in the Engineering Auditorium, 29 W. 39th St., New York, N. Y., on "New Things—Developments During the Depression."

As examples, a large number of new products, new machines, new materials and new devices were exhibited in the Engineering Auditorium during the address, and others were illustrated and described by the speaker.

"It is impossible to put the brakes on human ingenuity or stop the human mind from conceiving new ideas and evolving new inventions," said Dr. Parmelee. "The process can be accelerated under stress of necessity, but it can't be controlled by traffic signs. And that is what has happened during the past year or two of business depression. The demand for new products and new processes, for new materials and new uses, to stimulate sales, to cut costs, to establish new industries and revive old ones, has found expression in a wealth of invention and discovery. Science and engineering have again been mobilized to fight the battle against depression just as they were called to the colors during the war."

Among the new products referred to or shown during the address, were: talking signs; whirling boilers; air-conditioning outfits; cotton houses; filamentless radio tubes; copper sheeting; rubber textiles; teleregister for stock quotations; aluminum foil insulation; colored aluminum; glass bricks; metallized wood; runless stockings; crystal speakers; humidity indicators; typewriter that casts type; buses that run on rails or road; metal sprays; steel houses; sodium vapor lamps; linoleum wall coverings; high-speed snapshots; new foods; glass blackboards; heating screens; electronic telautograph; automaton bridge table; floating bricks; creaseless fabrics; vertical radio antennas.

MEETINGS

American Railway Engineering Association—Annual Meeting, Chicago, Ill., Mar. 15-17. E. H. Fritch, secretary, 59 E. Van Buren St., Chicago, Ill.

American Welding Society—Annual Meeting, New York, N. Y., Apr. 24-28.

Case School of Applied Science and The Cleveland Engineering Society—Joint conference on "Re-Engineering for Economical Manufacture," Cleveland, O., Mar. 22-24.

EXHIBITIONS

American Management Association — Third Annual Packaging, Packing and Shipping Exposition, Conference and Clinic, Hotel Pennsylvania, New York, N. Y., Mar. 7-10.

Knitting Arts Exhibition—Commercial Museum, Philadelphia, Pa., Apr. 24-28.

National Alliance of Art and Industry—Exhibition of "New Materials, New Products and New Uses," 65 E. 56th St., New York, N. Y., Jan. 16-Feb. 15.

National Association of Engine and Boat Manufacturers — Annual Power Boat Show, Grand Central Palace, New York, N. Y., Jan. 20-28.

National Automobile Show — Coliseum, Chicago, Ill., Jan. 28-Feb. 4.

New A.S.A. Standards

Approval has been granted by the American Standards Association to a new American standard on spur gear tooth form. This standard is a combination of the data on the 14½-deg. composite system and the 20-deg. stub involute system approved by A.S.A. in 1927 as American Tentative Standard (now promoted to the status of American Standard), together with new data referring to the 14½- and 20-deg. full-depth involute systems. The combined standard gives the detailed tooth proportions in terms of both the diametral pitch and the circular pitch for all four designs; and in addition, for the 14½-deg. and the 20-deg. full-depth involute systems, data for obtaining full involute action with pinions having small numbers of teeth are tabulated in convenient form.

A tentative national standard for abbreviations for the most common scientific engineering terms has been developed under the auspices of the A.S.A. and published by the American Society of Mechanical Engineers. This standard covers abbreviations ranging all the way from acres to kilovolt amperes.

NEW MATERIALS and PARTS

"Federalite" Alloy Bronze

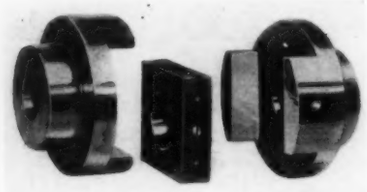
Is a modification of iron-bearing aluminum bronze, developed originally for use in motor-boat propellers where high-strength metal of easy flowing characteristics is necessary for thin blade sections. High corrosion-resistance claimed. Physical characteristics as determined on standard 0.505 in. test bars:

Tensile strength...80,000 lb. per sq.in.
Proportional limit..30,000 lb. per sq.in.
Elongation .....30 per cent
Hardness .....72 Rockwell B
Elastic modulus.....17,000,000

Federal-Mogul Corp., Shoemaker and Lillibridge Sts., Detroit, Mich.

D. O. James Flexible Coupling

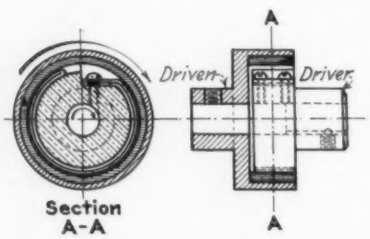
Composed of three parts: non-metallic center member, lubricated by self-contained reservoir, floats between jaws of one metal flange on line passing through center, and is free to slide across face



of second flange in direction at right angles to first. Claimed that flexibility is thus obtained without use of flexible material. Available for shafts ranging from $\frac{3}{4}$ to 13 in. diam. D. O. James Mfg. Co., 1114 W. Monroe St., Chicago, Ill.

Raybestos Thermal Clutch

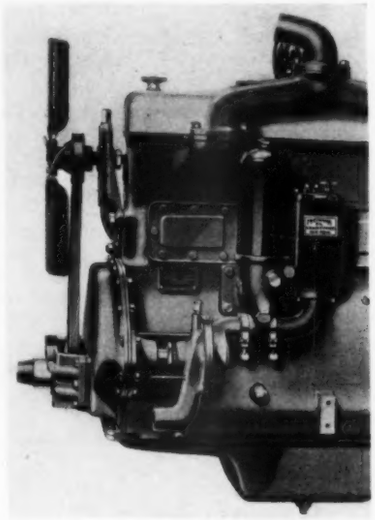
Adapted to small motor drives requiring gradual engagement. Has driving and driven surfaces between which torque is transmitted by expansion of bi-metallic thermostatic strip so designed as to release when driving member stops. Slippage at full torque



capacity and maximum motor speed claimed less than 5 per cent. By use of different types of bi-metal, period of delayed engagement may be extended to 3 seconds or shortened to 1/5 second as desired. The Raybestos Division, Raybestos-Manhattan, Inc., Bridgeport, Conn.

Michiana Oil Conditioner

Combines standard H-W oil filter and standard Harrison Viscon oil temperature control for direct crankcase application. Cooling unit is interposed



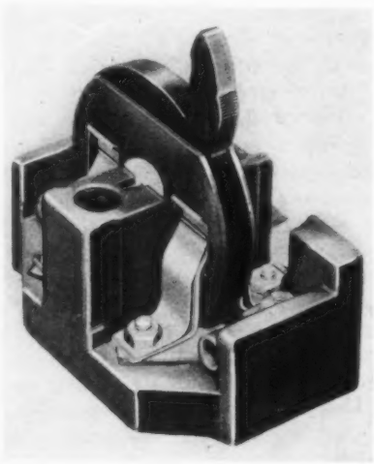
between radiator and engine. Cast-iron base forms sump to oil filter and also housing for oil temperature control unit. Filter available with either standard felt compression element or with "micro-mesh" screen element. Michiana Products Corp., Michigan City, Ind.

Manhattan "Compensated" Rubber Belting

Designed for low-tension operation. Pulley surface claimed to possess double coefficient of friction of standard rubber belting and to develop twice rated horsepower at one-half tension and with one-half amount of slip and creep. "Compensated" construction said to minmize working loose of fasteners through equalization of tension of outside duck ply with that of inside ply. Manhattan Rubber Mfg. Division, Raybestos-Manhattan, Inc., Passaic, N. J.

Square D Disk Switch

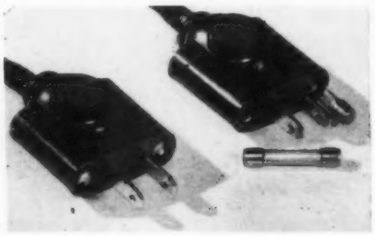
Has quick make and quick break action. May be obtained fused or unfused. Features listed by manufacturer:



front operation; small mounting space; copper contacts reinforced by rustless steel springs, insuring constant pressure and reduced heating. Rated for either 30 or 60 amp., single pole, two pole, two wire solid neutral or three wire solid neutral. Square D Co., Detroit, Mich.

El Menco Fused Attachment Plugs

Permit short circuits to be definitely localized to individual equipment. Fuses furnished in wide range of capacities.



Three-way plug also available. Electro-Motive Engineering Corp., 797 E. 140th St., New York, N. Y.

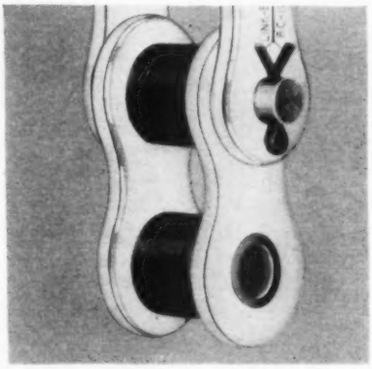
Leroy Lettering System

Consists of templets made of special laminated material with engraved letters showing black on white; fixed or adjustable scribers to produce either

vertical or slanting letters from same templet; and pens in ten sizes which can also be used for freehand lettering when fitted in special holders. Complete set of letters and numbers engraved on each standard templet, furnished in 12 sizes. Keuffel & Esser Co., Hoboken, N. J.

Link-Belt "Silverlink" Roller Chain

With sidebars treated to resist corrosion. Available in all sizes from $\frac{3}{8}$ to $2\frac{1}{2}$ in. pitch, and in single or multiple widths; also available with conveying



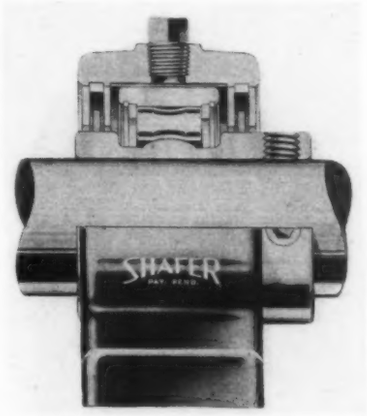
attachment links and as complete drives with sprockets. Features listed by manufacturer: heat-treated alloy steel sidebars; nickel steel casehardened pins; cotter pin or rivet fastenings; casehardened solid steel busings; heat-treated rollers of strip steel with fibre grain running around roller. Link-Belt Co., Indianapolis, Ind.

"Wearprooft" Localized Surface Hardening

Developed primarily for treating "MacHempIte" special alloy cast steel but applicable in varying degrees to other steels. Process claimed to introduce no stresses and no distortion requiring subsequent retruing. Either cast or machined surfaces may be treated to give hardnesses of from 60 to 90 Sclerescope. Mackintosh-Hemphill Co., Pittsburgh, Pa.

Shafer Light-Duty Roller Bearings

Available in flange units, hanger boxes and take-up units for shafts from $\frac{7}{8}$ to $2\frac{7}{8}$ in., in addition to pillow blocks as illustrated. Single row concave bearing (of full roller, cageless type) employed with rollers operating between straight outer race and convex inner race. Design said to assure liberal radial load capacity, with ample provision for limited thrust capacity needed



in moderate load applications. Self-alignment inherent in bearing itself; thrust taken on hardened and ground thrust plates. Shafer Bearing Corp., 621 S. Kolmar Ave., Chicago, Ill.

Stearns Compact Magnetic Clutch

Only 5 in. in diameter and $9\frac{1}{2}$ lb. in weight but developing 500 lb. in pull-out torque. Current consumption approximately 100 watts. Collector rings located on outside diameter, thus saving considerable space over usual location.

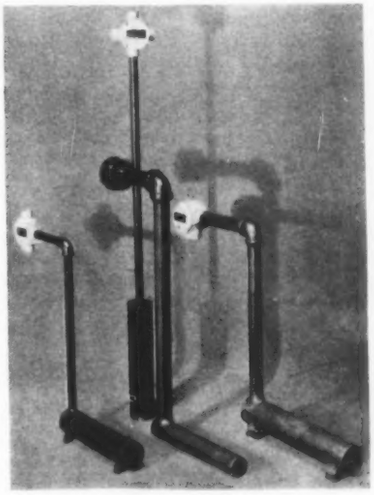
Clutch has only one friction or wearing surface and no moving parts. Magnetic Mfg. Co., Milwaukee, Wis.

Anaconda Beryllium Copper

Available in sheets, strips, rods, wire and tubes in sizes and gages in which phosphor bronze is ordinarily furnished. Also available in die-pressed forgings within certain manufacturing limitations. Alloy containing 2.25 per cent beryllium and balance copper has been adopted as standard. (See *P.E.*—September, 1932 for physical characteristics.) The American Brass Co., Waterbury, Conn.

Chromalox Heavy-Duty Electric Immersion Heaters

Built especially to consumer's requirements in steel or wrought-iron casing for salt baths, oil baths, and compound tanks; in cast-iron casing for metal melting pots for lead, solder, babbitt, etc. Each type built in ratings from 1.5 to 5 kw., either 115 or 230 volts. Edwin L. Wiegand Co., 7500 Thomas Blvd., Pittsburgh, Pa.

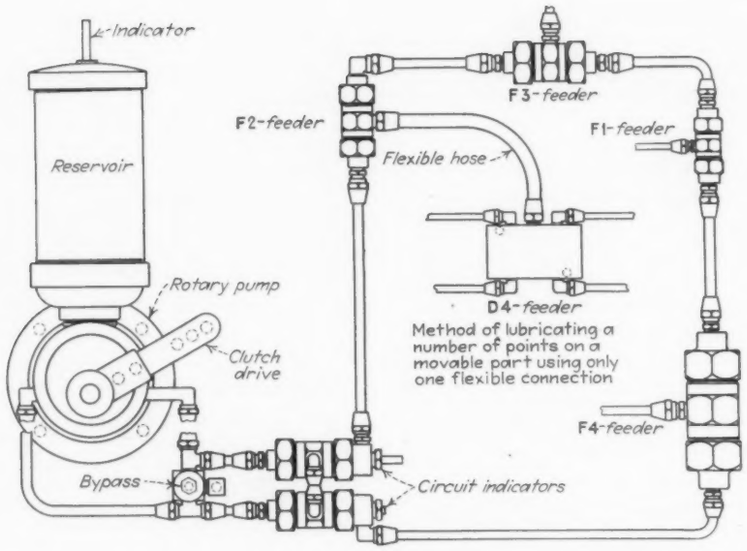


Goodrich Oil-Resistant Rubber Belting

Designed also to resist destructive action of certain acids and alkalies. Claimed that new belting has given satisfactory service after complete immersion in oil for several days and will last three times as long as ordinary rubber belting subjected to similar exposure. The B. F. Goodrich Co., Akron, Ohio.

Improved Trabon Lubricating System

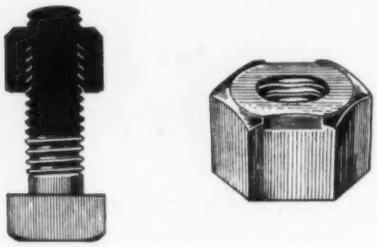
Designed to furnish automatic pressure lubrication to machine bearings through one-tube circuit with feeders. Forced-feed supplied either by small



motor driving rotary displacement pump, or by ratchet-connection of pump to some continuously moving part of machine being lubricated. Cycle of lubrication adjustable to any interval between 5 minutes and 8 hours. In typical installation, two circuit indicators are placed in line, one at head of circuit and other at its close. Lubricating Equipment Co., Detroit, Mich.

Bayonne "Extruded" Locking Nut

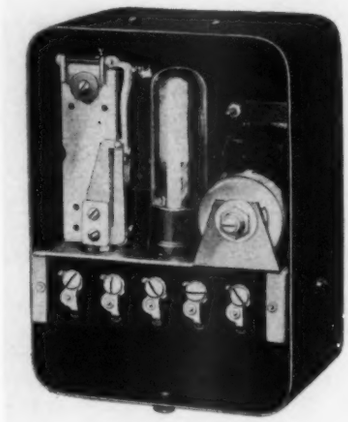
Depends for locking on deflected upper threads. May be taken off one bolt and placed on another without lessening its efficiency. Nut requires



wrench pressure for application and for release. Applicable to practically any standard bolt thread. Available in standard types and sizes from $\frac{1}{4}$ in. to $1\frac{1}{2}$ in. diameter, and on special order up to 2 in. Bayonne Bolt Corp., Bayonne, N. J.

General Electric CR7505-K1 Photoelectric Relay

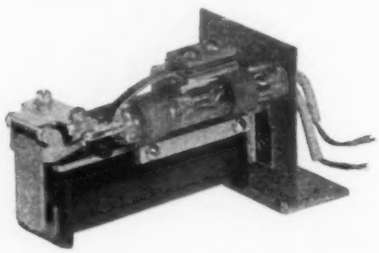
Designed for installations where practically complete interception of light beam is made, since changes of light intensity of 50 per cent are necessary for successful operation. Will operate on light impulses of 1/15 second duration. Unit consists of phototube, Pliotron amplifying tube, sensitive relay, adjusta-



able potentiometer and associated apparatus all mounted on steel base. Drawn shell inclosing case provided. General Electric Co., Schenectady, N. Y.

"Automatic" Vacuum Contacting A. C. Relay

Of standard design long used in automatic telephone apparatus but modified to accommodate vacuum contactor. Opens and closes circuit carrying up to



6 amp. at 110 or 220 volts. Break takes place in vacuum thus preventing danger of ignition of gases, dust, etc. Automatic Electric Co., 1033 W. Van Buren St., Chicago, Ill.

Lukens Improved Nickel-Clad Steel

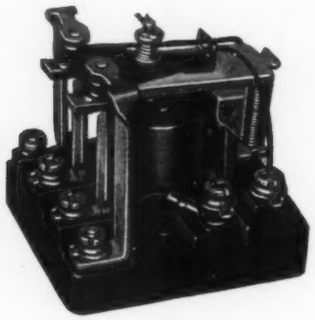
Available in new "hot-rolled and cleaned" finish free from brown nickel oxide on nickel side. Available in thicknesses of $\frac{1}{16}$ to 1 in., in plates ranging from 24x78 to 144x200 in. Nickel layers may be had in thicknesses up to 20 per cent of total plate thickness. Lukens Steel Co., Coatesville, Pa.

No. 2886 "Mars-Lumograph" Pencils

Are claimed to obviate necessity of India ink in production of drawings and tracings for blueprints, lines drawn with these pencils being impenetrable to light. Adaptable to any kind of pencil work. Obtainable in 7 deg. of hardness, Ex B, HB, F, H, 2H, 3H and 4H. J. S. Staedtler, Inc., 53-55 Worth St., New York, N. Y.

Ward Leonard Midget Magnetic Relay

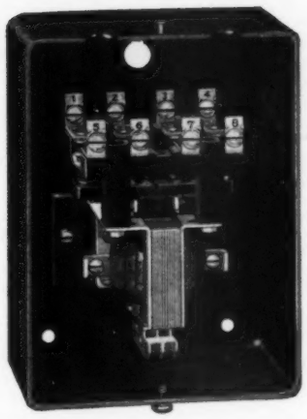
For remote control. New molded base provides more rigid mounting of contacts and binding posts. Newly designed armature said to improve magnetic circuit and alignment of contacts with reduced energy consumption. Available in single and double pole,



single and double throw and various arrangements of contacts and terminal connections for a.c. and d.c. up to 110 volts. Ward Leonard Electric Co., Mount Vernon, N. Y.

Allen-Bradley Bulletin 700 Solenoid Contactors

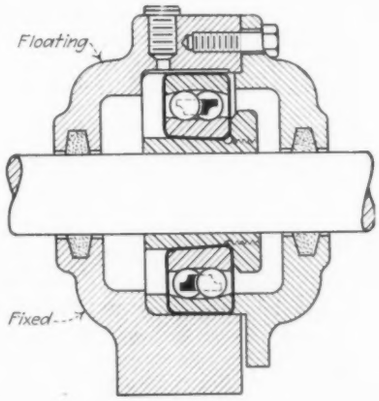
Supplied for a.c. applications requiring remote control switch. Available in single or multiple pole construction with silver-to-silver contacts up to 8-pole unit for 2-wire control. All contacts being supported on self-contained insulating cross pieces, contactors can be mounted directly on metal panels or machines. Standard voltages 6, 12, 24,



110 and 220; amperes 5 to 30, depending upon type and voltage. Allen-Bradley Co., 1311 S. First St., Milwaukee, Wis.

Norma-Hoffman Ball Bearing Pedestal

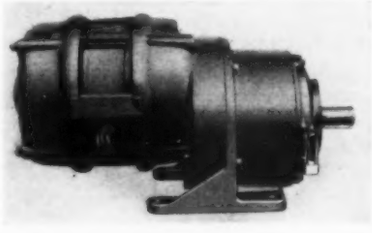
Designed as unit mounting for pillow blocks or for direct machine application. Designated as "Lup" series.



Available in sizes to fit standard shafting in nominal inch as well as sixteenth diameters from $\frac{15}{16}$ to $3\frac{1}{2}$ in. Ball bearings are standard "Precision" double-row, self-aligning, with adapter sleeve. Felt lubricant-seals. Norma-Hoffman Bearings Corp., Stamford, Conn.

"Ideal" Geared Motors

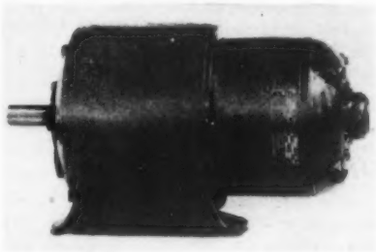
With high-speed gear mounted directly upon motor shaft. Available in all gear ratios from 2.09:1 to 376:1 giving output speeds of from 861 r.p.m.



to 4.7 r.p.m. with standard 1,800 r.p.m. motor. Helical gears or combination of helical and heliocentric gears run in oil bath. Ball or roller bearings. Units employing heliocentric gears have motor hung on reducer as shown in illustration. Available in horsepowers from $\frac{3}{4}$ to 20. Motors either open or inclosed, for 2 or 3 phase a.c., or for d.c. Two, three, or four speed motors also available. The Ideal Electric & Mfg. Co., Mansfield, Ohio.

Louis Allis Motorized Speed Reducers

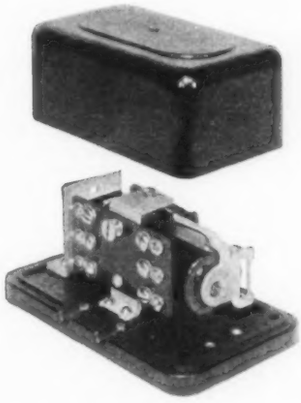
Are available in ratings from $\frac{3}{4}$ to 75 hp., with output speed 4 to 400 r.p.m. Built in styles accommodating nearly any type of motor. Integral style shown in illustration can be furnished where an open or inclosed non-ventilated



motor is employed. Manufacturer claims high efficiency through use of precision gears and oversized ball bearings. The Louis Allis Co., Milwaukee, Wis.

"Automatic" Relay Mounting Box

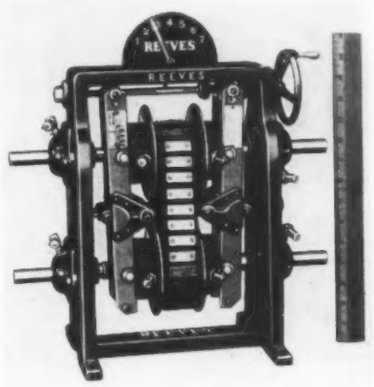
For almost any type small relay. Consists of two parts, base and cover, both made of Bakelite. Base drilled with holes for screw mounting. Small terminal



strip, arranged to mount integrally with relay on inside of box, facilitates making connections. Dimensions: 3 in. wide, $5\frac{1}{8}$ in. long; $2\frac{1}{4}$ in. high. Automatic Electric Co., 1033 W. Van Buren St., Chicago, Ill.

Reeves No. 0000 Variable-Speed Transmission

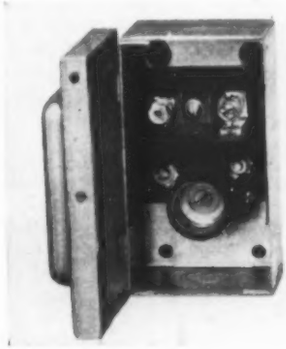
Is claimed to be the smallest vertical design, variable-speed transmission. Variable-speed range from 2:1 to 6:1



inclusive; hp. capacity from $\frac{1}{8}$ to $\frac{3}{4}$. Available also in completely inclosed design or equipped with motor base and individual driving motor with auxiliary countershaft for greater speed reduction. Reeves Pulley Co., Columbus, Ind.

Square D No. 90211 Disk Switch

Is two-pole switch, $5\frac{3}{8}$ in. high by $3\frac{1}{8}$ in. wide by $2\frac{3}{4}$ in. deep, with one fuse, one blade and solid neutral. Capacity 30 amp. or 1 hp. at 125 volts. Disk feature permits small size cabinet



and convenient front operation, without necessity of providing for blade swing or side-operating handle. Generous wiring space. Square D Co., Detroit, Mich.

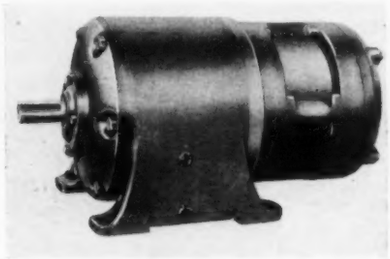
Trumbull No. 68361 Switches

In 30, 400, 600 and 800-amp. sizes, 2, 3 and 4-pole, 230 and 575 volts a.c. both side and front operated are additions to the "R.B." line of industrial switches. Feature "Vystipe" fuse clamping device claimed to reduce heating at fuse terminals, and a new type of roll contact designed to main-

tain line contact under heavy pressure. The Trumbull Electric Mfg. Co., Plainville, Conn.

Allis-Chalmers Motor-Reduction Units

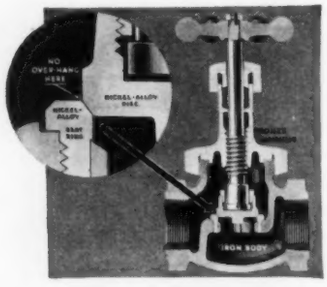
For low-speed drive. Can be supplied with any desired type of motor. Helical gears; ball or roller bearings; automatic lubrication. Standard speeds at



power take-off are from 3.02 r.p.m. to 380 r.p.m., but lower speeds can be provided. Available for floor, wall, ceiling or vertical mounting. Allis-Chalmers Mfg. Co., Milwaukee, Wis.

Jenkins Fig. 890 Globe Valves

Supplied with renewable and regrindable nickel alloy, bronze or stainless steel seat rings and disks. Because disk does not overhang bevel contact sur-



face of seat ring, this iron body bronze trimmed valve can be reground repeatedly without danger of cutting shoulder into disk. Manganese bronze spindle. Available in $\frac{3}{8}$ to 2 in. sizes. Jenkins Bros., 80 White St., New York, N. Y.

NEW BOOKS and PUBLICATIONS

Research Bulletins

American Society for Testing Materials, 1315 Spruce St., Philadelphia, Pa. "1932 A.S.T.M. Proceedings" in two parts. Part I, 1071 pages, contains the annual reports of the many Society committees and the technical papers and standards appended thereto. Part II, 825 pages, includes the many technical papers presented at the annual meeting and discussions thereon. Prices for each part, $5.50 paper binding, $6.00 cloth, $7.00 half leather. "Index to A.S.T.M. Standards and Tentative Standards," 114 pages. Copies may be obtained without charge on application to Society headquarters.

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Bearings—Merriman Bros., Inc., 185 Amory St., Jamaica Plain, Boston, Mass. Bulletin No. 33, "Lubrite Bushings and Bearings."

Beryllium Copper — The American Brass Co., Waterbury, Conn. Publication B-21, "Anaconda Beryllium Copper a Heat-Treatable Copper Alloy."

Conveyor Belting—The B. F. Goodrich Rubber Co., Akron, Ohio. Catalog insert, Maxecon general service conveyor belt and Goodrich improved hot material belt.

Electric Heaters—Edwin L. Wiegand Co., 7500 Thomas Blvd., Pittsburgh, Pa. Catalog sheet C-115-2, "Chromalox Heavy-Duty Electric Immersion Heater."

Expansion Joints — Yarnall-Waring Co., Chestnut Hill, Philadelphia, Pa. Bulletin EJ-1904, "Yarway Cylinder-Guided Expansion Joints."

Glue—Tego Gluefilm, Inc., Terre Haute, Ind. Booklet on sheeted glue for laminated products.

Leather Belting—E. F. Houghton & Co., Philadelphia, Pa. Booklet, "Vim Tred Leather Belting."

Lettering — Keuffel & Esser Co., Hoboken, N. J. Folder, "The LeRoy Lettering System."

Micro Analysis—E. Leitz, Inc., 60 E. 10th St., New York, N. Y. Bulletin No. 2, "Darkfield Microscopy for Micro Analysis of Metals"; Bulletin No. 3, "Leitz Micro-Metallograph Simplified Model MM-2."

Monel Metal — The International Nickel Co., Inc., 67 Wall St., New York, N. Y. Bulletin T-5, "Engineering Properties of Monel Metal."

Motors — Reliance Electric & Engineering Co., Cleveland, Ohio. Two bulletins: No. 114, "Low-Inertia Special-Torque A-C Motors"; No. 116, "Form OSH Type AA Reliance Motors."

Roller Chains—Link-Belt Co., Indianapolis, Ind. Data Book 1457, "Link-Belt Roller Chains."

Special Equipment—The Patterson Foundry & Machine Co., East Liverpool, Ohio. Brochure, "Special Processing Equipment."

Springs—Sleeper & Hartley, Inc., Worcester, Mass. Catalog of spring-making machinery, etc.

V-Belts—The B. F. Goodrich Rubber Co., Akron, Ohio. Insert No. 2182 for mechanical goods catalog, "Longlife V-belts."

"Man and Metals"

A history of mining in relation to the development of civilization

T. A. Rickard. 2 volumes, 1068 pages, 6x9 in. Illustrated. Published by Whittlesey House, McGraw-Hill Book Company, Inc., New York, N. Y. Price $10.

In times like these when each one of us is baffled by his immediate problems and environment, and when the future seems all too dark, Mr. Rickard's book comes as a welcome means of restoring one's perspective. Inspired by Wells' Outline of History, partly because he thinks Wells paid insufficient attention to the part played by the miner in man's development, Mr. Rickard has set out to fill this gap. His many years of experience in technical journalism, as editor of *Engineering and Mining Journal, Mining and Scientific Press,* and *The Mining Magazine,* it is very evident, have stood him in good stead in this present work. These two volumes are not only well done from the technical point of view; they are also very entertaining reading.

Mr. Rickard makes it clear that the powerful nations in history have been those that developed or controlled mineral resources. He tells how the Athenians used the silver from their mines to finance the building of the ships with which they repelled the Persians; and how the gold of California enabled the North to wear down the poorly financed South in our own Civil War. And he makes it plain that the United States of America boasts the most lavish endowment of mineral resources ever known, ample insurance of its future growth and power in the world, which should certainly be comforting to our present doubts.

And his account of the struggle of the miner up from the most abject slavery of classic times to a position of unique freedom is a thrilling epic. All in all, "Man and Metals" is well worth the time it takes to read it. It should prove inspiring to every man who has a part to play in the modern metal industry.

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Horsepower Capacities of Spur Gears II

(PER INCH OF FACE WIDTH)

A complete explanation of the accompanying tables of horsepower ratings of spur gears is given on page 39 of the January *Product Engineering*. The tables are for the A.G.M.A. standard 14½ deg. pressure angle involute cast-iron spur gears. Multiplying factors for the 20-deg. full depth tooth and the 20-deg. stub tooth are given in the above-mentioned article, which also contains a table for multiplying factors to be used when the gear material is other than cast iron.

In the selection of the gear material, when the drive ratio is 1 to 1, the driven and driving members can be of the same material, although better operation is obtainable if one of the gears is of a more ductile material. Because the strength of the gear tooth decreases with a smaller number of teeth, in pinion and gear combination the pinion is usually made of steel or a material stronger than that used for the gear. The latter may be made of cast iron, semi-steel, or cast steel. This is not a fixed rule as cast iron is frequently employed for both pinion and gear with excellent results.

For extreme service conditions, heat-treated or casehardened alloy steel pinions are usually used, the mating gear being of cast or forged steel in either the normalized or the heat-treated condition.

As an example of the use of the tables, assume it is desired to transmit 10 hp. at an input speed of 1200 r.p.m., and that the gear ratio is to be 4.200 to one. The pinion is to be of 1045 heat-treated steel cut solid on its shaft, and the gear is to be of forged steel. The drive is to operate in an oil bath; the pitch, face, and center distance to be kept to a minimum.

Referring to the material tables (Jan. *P. E.*, page 39), the multiplying factor for this pinion material is 3.750. Dividing this factor into the specified hp. of 10, it is found that an equivalent cast-iron pinion must be capable of transmitting 0.667 hp. at 1200 r.p.m.

Considering the gear, the material factor for forged steel is 2.000, indicating that the gear will be the weaker member. The equivalent cast-iron gear must then be able to transmit 10/2=5 hp. at the gear speed which is namely, 1200 divided by the speed ratio of 4.200, or 286 r.p.m. Since it is always well to use a pinion of 15 or more teeth at the higher speeds, a 15-tooth pinion will be used. The gear will then have 15x4½=63 teeth. From a casual inspection it is found that neither 12 D.P. nor 10 D.P. are strong enough but that 8 D.P. probably will be. The equivalent cast-iron gear of 63 teeth, 8 D.P. at 200 r.p.m. will transmit 2.64 hp. per inch of face width, as found by interpolating in the table of horsepower capacity for 8 D.P. Similarly, at 300 r.p.m., the 63 tooth gear will transmit 3.28 hp. per inch of face width. Therefore, at 286 r.p.m. this gear will transmit, as found by interpolation, 3.13 hp. per in. of face width. Dividing 5.00 (the equivalent hp. to be transmitted) by 3.13, the gear face width will be 1.6 in., or 1⅝ in. This is somewhat wider than usual, but may be acceptable. The gear will then transmit 3.13×1.625×2.0=10.17 hp.

The 15-tooth, 8 D.P. pinion with 1⅝ in. face width will transmit, as determined from the tables, 2.22×1.625×3.750=13.5 hp.

Contributed by C. H. Grill

4—Diametral Pitch

R.P.M.		Number of Teeth								
		12	14	16	18	20	25	30	50	100
1800	V.	1415	1650	1890	2120	2360	....	....	....	....
	H.P.	5.4	6.3	7.1	7.7	8.2	....	....	....	....
1200	V.	945	1100	1260	1415	1570	1965	2360	....	....
	H.P.	4.8	5.6	6.3	6.9	7.4	8.5	9.2	....	....
900	V.	710	825	945	1060	1180	1475	1770	....	....
	H.P.	4.2	5.0	5.7	6.3	6.8	7.9	8.6	....	....
600	V.	470	550	630	705	785	985	1180	1965	....
	H.P.	3.45	4.1	4.8	5.4	5.9	6.9	7.7	9.6	....
450	V.	355	410	470	530	590	740	885	1475	....
	H.P.	2.90	3.50	4.1	4.6	5.1	6.1	6.9	8.9	....
300	V.	235	275	315	355	395	490	590	985	1965
	H.P.	2.20	2.70	3.20	3.65	4.1	5.0	5.7	7.8	10.3
200	V.	157	183	210	235	260	330	395	655	1310
	H.P.	1.61	2.00	2.44	2.76	3.10	3.95	4.6	6.6	9.2
150	V.	118	137	157	177	197	245	295	490	985
	H.P.	1.28	1.60	1.94	2.24	2.54	3.20	3.80	5.7	8.3
100	V.	79	92	105	118	131	164	196	330	655
	H.P.	.88	1.13	1.41	1.61	1.84	2.38	2.84	4.5	7.0
50	V.	39.0	45.5	52	59	66	82	98	164	330
	H.P.	.44	.56	.70	.83	.96	1.30	1.62	2.70	4.8
10	V.	7.9	9.2	10.5	11.8	13.1	16.4	19.6	33.0	66
	H.P.	.088	.113	.140	.167	.191	.260	.320	.59	1.26
1	V.	....	....	1.05	1.18	1.31	1.64	1.96	3.3	6.6
	H.P.	....	....	.014	.017	.019	.026	.032	.059	.13

5—Diametral Pitch

R.P.M.		Number of Teeth								
		12	14	16	18	20	25	30	50	100
1800	V.	1130	1320	1510	1700	1885	2360	....	....	....
	H.P.	4.1	4.7	5.4	5.8	6.2	7.1	....	....	....
1200	V.	755	880	1010	1130	1260	1570	1885	....	....
	H.P.	3.50	4.1	4.7	5.1	5.5	6.5	6.9	....	....
900	V.	565	660	755	850	945	1180	1415	2360	....
	H.P.	3.05	3.60	4.2	4.6	5.0	5.9	6.4	8.0	....
600	V.	380	440	505	565	630	785	945	1570	....
	H.P.	2.42	2.90	3.40	3.80	4.2	5.0	5.6	7.3	....
450	V.	285	330	380	425	470	590	710	1180	2360
	H.P.	2.02	2.44	2.90	3.25	3.60	4.4	5.0	6.7	8.5
300	V.	189	220	250	285	315	390	470	785	1570
	H.P.	1.50	1.84	2.20	2.53	2.82	3.50	4.1	5.7	7.7
200	V.	126	147	168	188	210	260	315	525	1050
	H.P.	1.08	1.35	1.64	1.87	2.14	2.68	3.20	4.7	6.8
150	V.	94	110	126	141	157	196	235	390	785
	H.P.	.84	1.08	1.30	1.50	1.70	2.18	2.60	3.90	6.1
100	V.	63	73	84	94	105	131	157	260	525
	H.P.	.56	.73	.90	1.05	1.23	1.59	1.88	3.00	5.0
50	V.	31.5	36.5	42.0	47.0	52	66	79	131	260
	H.P.	.280	.365	.45	.53	.62	.83	1.04	1.79	3.25
10	V.	6.3	7.3	8.4	9.4	10.5	13.1	15.7	26.2	52
	H.P.	.056	.073	.090	.105	.123	.166	.208	.375	.80
1	V.	....	....	....	....	1.05	1.31	1.57	2.6	5.2
	H.P.	....	....	....	....	.012	.017	.021	.037	.080

6—Diametral Pitch

R.P.M.		Number of Teeth								
		12	14	16	18	20	25	30	50	100
3600	V.	1880	2200	....	....	....	....	....	....	....
	H.P.	3.95	4.5	....	....	....	....	....	....	....
1800	V.	945	1100	1260	1415	1570	1970	2360	....	....
	H.P.	3.20	3.70	4.2	4.6	5.0	5.7	6.1	....	....
1200	V.	630	735	840	945	1050	1310	1570	....	....
	H.P.	2.66	3.15	3.65	4.0	4.4	5.1	5.6	....	....
900	V.	470	550	630	710	785	985	1180	1970	....
	H.P.	2.28	2.74	3.20	3.55	3.90	4.6	5.1	6.4	....
600	V.	315	365	420	470	525	655	785	1310	....
	H.P.	1.79	2.16	2.58	2.88	3.20	3.85	4.4	5.7	....
300	V.	157	183	210	235	260	330	390	655	1310
	H.P.	1.08	1.34	1.62	1.84	2.06	2.62	3.05	4.4	6.1
200	V.	105	122	140	157	175	220	260	435	875
	H.P.	.78	.96	1.18	1.36	1.54	1.98	2.33	3.50	5.3
150	V.	79	92	105	118	131	164	197	330	655
	H.P.	.59	.75	.94	1.07	1.23	1.58	1.89	2.97	4.6
100	V.	52	61	70	79	87	109	131	220	435
	H.P.	.390	.50	.63	.74	.85	1.15	1.38	2.24	3.75
50	V.	26.2	30.5	35.0	39.0	43.5	55	66	109	220
	H.P.	.196	.250	.310	.370	.43	.58	.72	1.31	2.40
10	V.	5.2	6.1	7.0	7.9	8.7	10.9	13.1	21.8	43.5
	H.P.	.039	.050	.063	.074	.085	.115	.144	.264	.55
1	V.	....	....	....	....	....	1.09	1.31	2.2	4.4
	H.P.	....	....	....	....	....	.011	.015	.026	.055

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REFERENCE BOOK SHEET

Horsepower Capacities of Spur Gears—II

(PER INCH OF FACE WIDTH)

7—Diametral Pitch

R.P.M.		Number of Teeth								
		12	14	16	18	20	25	30	50	100
3600	V.	1620	1890	2160	.....	.....	.....	.....	.....	.....
	H.P.	3.25	3.70	4.2	.....	.....	.....	.....	.....	.....
1800	V.	810	945	1080	1210	1350	1680	2020	.....	.....
	H.P.	2.56	3.00	3.45	3.75	4.1	4.7	5.1	.....	.....
1200	V.	540	630	720	810	900	1120	1350	2240	.....
	H.P.	2.12	2.50	2.92	3.20	3.50	4.1	4.6	5.7	.....
900	V.	405	470	540	605	675	840	1010	1680	.....
	H.P.	1.80	2.15	2.54	2.82	3.10	3.70	4.2	5.3	.....
600	V.	270	315	360	405	450	560	675	1120	2250
	H.P.	1.40	1.69	2.00	2.26	2.52	3.05	3.50	4.7	6.1
300	V.	135	157	180	200	225	280	335	560	1120
	H.P.	.83	1.01	1.24	1.41	1.60	2.02	2.36	3.45	5.0
200	V.	90	105	120	135	150	187	225	375	750
	H.P.	.57	.73	.89	1.03	1.18	1.50	1.80	2.78	4.2
150	V.	67	79	90	101	112	140	168	280	560
	H.P.	.43	.55	.69	.81	.92	1.20	1.44	2.28	3.70
100	V.	45.0	52	60	67	75	94	112	187	375
	H.P.	.288	.370	.46	.54	.63	.85	1.05	1.70	2.94
50	V.	22.5	26.2	30.0	33.5	37.5	47.0	56	94	187
	H.P.	.144	.184	.230	.270	.315	.42	.53	.96	1.81
10	V.	4.5	5.2	6.0	6.7	7.5	9.4	11.2	18.7	37.5
	H.P.	.029	.037	.046	.054	.063	.085	.105	.193	.41
1	V.	.....	.....	.....	.....	.....	.....	1.12	1.87	3.7
	H.P.	.....	.....	.....	.....	.....	.....	.010	.019	.041

12—Diametral Pitch

R.P.M.		Number of Teeth								
		12	14	16	18	20	25	30	50	100
3600	V.	945	1100	1260	1410	1570	1970	2360	.....	.....
	H.P.	1.59	1.85	2.12	2.30	2.48	2.84	3.05	.....	.....
2400	V.	630	735	840	945	1050	1310	1570	.....	.....
	H.P.	1.33	1.57	1.83	2.00	2.18	2.54	2.78	.....	.....
1800	V.	470	550	630	705	785	985	1180	1970	.....
	H.P.	1.14	1.37	1.60	1.77	1.94	2.30	2.55	3.20	.....
1200	V.	315	365	420	470	525	655	785	1310	.....
	H.P.	.90	1.08	1.29	1.44	1.60	1.93	2.18	2.88	.....
900	V.	235	275	315	355	390	490	590	985	1970
	H.P.	.73	.90	1.08	1.22	1.35	1.66	1.91	2.60	3.40
600	V.	157	183	210	235	260	330	390	655	1310
	H.P.	.54	.67	.81	.92	1.04	1.30	1.53	2.18	3.05
300	V.	79	92	105	118	131	163	197	330	655
	H.P.	.294	.375	.47	.54	.61	.79	.95	1.49	2.32
200	V.	52	61	70	79	87	109	131	220	435
	H.P.	.196	.250	.310	.370	.43	.57	.69	1.12	1.87
150	V.	39.0	46.0	52	59	66	82	98	164	330
	H.P.	.147	.188	.234	.276	.320	.43	.54	.89	1.58
100	V.	26.2	30.5	35.0	39.0	43.5	55	66	109	220
	H.P.	.098	.125	.156	.184	.212	.288	.360	.65	1.20
50	V.	13.1	15.3	17.5	19.7	21.8	27.3	33.0	55	109
	H.P.	.049	.062	.078	.092	.107	.144	.180	.325	.69
10	V.	.....	.....	.....	.....	.....	.....	.....	10.9	21.8
	H.P.	.....	.....	.....	.....	.....	.....	.....	.065	.139

8—Diametral Pitch

R.P.M.		Number of Teeth								
		12	14	16	18	20	25	30	50	100
3600	V.	1410	1650	1890	2120	2360	.....	.....	.....	.....
	H.P.	2.76	3.15	3.55	3.80	4.1	.....	.....	.....	.....
2400	V.	945	1100	1260	1410	1570	1960	2360	.....	.....
	H.P.	2.39	2.78	3.15	3.45	3.70	4.2	4.6	.....	.....
1800	V.	710	825	945	1060	1180	1470	1770	.....	.....
	H.P.	2.12	2.48	2.86	3.15	3.40	3.95	4.3	.....	.....
1200	V.	470	550	630	710	785	985	1180	1960	.....
	H.P.	1.71	2.05	2.40	2.66	2.90	3.45	3.80	4.8	.....
900	V.	355	415	470	530	590	735	885	1470	.....
	H.P.	1.45	1.75	2.06	2.30	2.54	3.05	3.45	4.5	.....
600	V.	235	275	315	355	395	490	590	985	1970
	H.P.	1.10	1.35	1.61	1.82	2.04	2.48	2.86	3.90	5.1
300	V.	118	138	157	177	197	245	295	490	985
	H.P.	.64	.80	.97	1.12	1.27	1.60	1.90	2.82	4.2
200	V.	79	92	105	118	131	164	197	330	655
	H.P.	.44	.56	.70	.81	.92	1.19	1.42	2.24	3.50
150	V.	59	69	79	89	98	123	147	245	490
	H.P.	.330	.42	.53	.62	.72	.94	1.14	1.82	3.00
100	V.	39.5	46.0	52	59	66	82	98	164	330
	H.P.	.220	.282	.350	.42	.48	.65	.81	1.35	2.38
50	V.	19.7	23.0	26.0	29.5	33.0	41.0	49.0	82	164
	H.P.	.110	.141	.176	.208	.240	.325	.40	.74	1.46
10	V.	.....	.....	.....	.....	.....	.....	9.8	16.4	33.0
	H.P.	.....	.....	.....	.....	.....	.....	.081	.148	.320

14—Diametral Pitch

R.P.M.		Number of Teeth								
		12	14	16	18	20	25	30	50	100
3600	V.	810	945	1080	1210	1350	1680	2020	.....	.....
	H.P.	1.28	1.50	1.72	1.88	2.04	2.34	2.54	.....	.....
2400	V.	540	630	720	810	900	1120	1350	2240	.....
	H.P.	1.06	1.25	1.46	1.61	1.76	2.06	2.28	2.84	.....
1800	V.	405	470	540	605	675	840	1010	1680	.....
	H.P.	.90	1.08	1.27	1.40	1.55	1.85	2.08	2.66	.....
1200	V.	270	315	360	405	450	560	675	1120	2240
	H.P.	.69	.84	1.00	1.12	1.26	1.53	1.75	2.34	3.00
900	V.	200	235	270	300	335	420	505	840	1680
	H.P.	.56	.69	.83	.93	1.05	1.31	1.51	2.10	2.80
600	V.	135	157	180	200	225	280	335	560	1120
	H.P.	.41	.51	.61	.70	.80	1.01	1.19	1.73	2.48
300	V.	67	79	90	101	112	140	168	280	560
	H.P.	.214	.276	.345	.40	.46	.60	.72	1.14	1.85
200	V.	45.0	52	60	67	75	94	112	187	375
	H.P.	.143	.184	.230	.268	.315	.43	.52	.85	1.47
150	V.	33.5	39.0	45.0	51	56	70	84	140	280
	H.P.	.107	.138	.172	.201	.236	.320	.395	.68	1.22
100	V.	22.5	26.2	30.0	34.0	37.5	47.0	56	94	187
	H.P.	.072	.092	.115	.134	.157	.212	.264	.48	.91
50	V.	11.2	13.1	15.0	16.8	18.7	23.4	28.0	47.0	94
	H.P.	.036	.046	.057	.067	.079	.106	.132	.241	.51
10	V.	.....	.....	.....	.....	.....	.....	.....	.....	18.7
	H.P.	.....	.....	.....	.....	.....	.....	.....	.....	.103

10—Diametral Pitch

R.P.M.		Number of Teeth								
		12	14	16	18	20	25	30	50	100
3600	V.	1130	1320	1510	1700	1890	2360	.....	.....	.....
	H.P.	2.04	2.36	2.68	2.90	3.10	3.55	.....	.....	.....
2400	V.	755	880	1000	1130	1260	1570	1890	.....	.....
	H.P.	1.74	2.04	2.34	2.57	2.78	3.20	3.50	.....	.....
1800	V.	565	660	755	850	945	1180	1410	2360	.....
	H.P.	1.51	1.80	2.08	2.30	2.51	2.94	3.25	4.0	.....
1200	V.	380	440	500	565	630	785	945	1570	.....
	H.P.	1.21	1.45	1.70	1.91	2.10	2.52	2.82	3.65	.....
900	V.	280	330	380	425	470	590	705	1180	2360
	H.P.	.99	1.22	1.45	1.63	1.81	2.20	2.50	3.35	4.3
600	V.	189	220	250	285	315	395	470	785	1570
	H.P.	.75	.92	1.10	1.26	1.41	1.76	2.02	2.86	3.85
300	V.	94	110	126	141	157	197	235	395	785
	H.P.	.42	.54	.65	.75	.85	1.09	1.30	1.98	3.05
200	V.	63	73	84	94	105	131	157	260	525
	H.P.	.280	.360	.45	.53	.62	.79	.96	1.52	2.50
150	V.	47.0	55	63	71	79	98	118	197	395
	H.P.	.210	.268	.340	.395	.46	.62	.76	1.24	2.12
100	V.	31.5	37.0	42.0	47.0	53	66	79	131	260
	H.P.	.140	.179	.225	.264	.310	.41	.52	.90	1.62
50	V.	15.7	18.3	21.0	23.6	26.2	33.0	39.5	65	131
	H.P.	.070	.090	.113	.132	.155	.206	.260	.47	.96
10	V.	.....	.....	.....	.....	.....	.....	.....	13.1	26.2
	H.P.	.....	.....	.....	.....	.....	.....	.....	.094	.199

16—Diametral Pitch

R.P.M.		Number of Teeth								
		12	14	16	18	20	25	30	50	100
3600	V.	705	825	945	1060	1180	1470	1770	.....	.....
	H.P.	1.06	1.24	1.43	1.57	1.70	1.97	2.18	.....	.....
2400	V.	470	550	630	705	785	980	1180	1960	.....
	H.P.	.86	1.02	1.20	1.32	1.46	1.72	1.92	2.40	.....
1800	V.	355	415	470	530	590	735	885	1470	.....
	H.P.	.73	.88	1.03	1.15	1.27	1.52	1.72	2.24	.....
1200	V.	235	275	315	355	390	490	590	980	1960
	H.P.	.55	.67	.81	.91	1.01	1.24	1.43	1.95	2.56
900	V.	177	205	235	265	295	370	440	735	1470
	H.P.	.44	.55	.66	.75	.85	1.06	1.22	1.73	2.38
600	V.	118	138	157	177	197	245	295	490	985
	H.P.	.320	.40	.49	.56	.64	.80	.95	1.41	2.08
300	V.	59	69	79	89	98	123	147	245	490
	H.P.	.163	.210	.264	.310	.360	.47	.57	.91	1.50
200	V.	39.0	46.0	52	59	66	82	98	164	330
	H.P.	.109	.140	.176	.208	.240	.325	.40	.68	1.19
150	V.	29.4	34.5	39.0	44.0	49.0	61	74	123	245
	H.P.	.082	.105	.132	1.56	.180	.243	.300	.54	.97
100	V.	19.7	23.0	26.2	29.4	33.0	41.0	49.0	82	164
	H.P.	.054	.070	.088	.104	.120	.162	.202	.370	.72
50	V.	9.8	11.5	13.1	14.7	16.4	20.5	24.6	41.0	82
	H.P.	.027	.035	.044	.052	.060	.081	.101	.184	.390
10	V.	.....	.....	.....	.....	.....	.....	.....	.....	16.4
	H.P.	.....	.....	.....	.....	.....	.....	.....	.....	.078

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